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XIV.

AN ACCOUNT OF THE EXPERIMENTS MADE ON THE DOUBLE-SCREW STEEL FERRY BOAT *BERGEN*, PLYING BETWEEN THE CITIES OF NEW YORK AND JERSEY CITY.

By Chief Engineer Isherwood, U. S. Navy.

The *Bergen* is a steel ferry boat with the peculiarity of being fitted with duplicate screws, one screw at each end and both upon the same shaft, which is, of course, operated by the same engine. The greatest immersed transverse section of the vessel is at exactly the midlength of the hull, and the two bodies, one on each side of this section, are precisely alike. The vessel is never turned, but runs alternately forwards and back, entering what are called "slips" at the ends of the route, and making the embarkations and discharges of passengers and vehicles from the ends of the boat. This method of using the boat compels the employment of a duplicate screw at each end, if screw propulsion be adopted, each screw exercising alternately a pushing and pulling effect upon the hull, and both acting simultaneously. The route is a short one of from seven to ten minutes' duration,

being the breadth of the Hudson River from the western end of Barclay street, in New York City, to Hoboken, in Jersey City. The *Bergen* is the first steamer in the United States propelled with what are termed "double screws," and marks an extreme innovation in the mode of propelling ferry boats, which heretofore has universally been done by the ordinary paddle wheel.

In England the *Bergen* was preceded by the *Oxton*, a ferry boat plying across the river Mersey, between Liverpool and Birkenhead, with twin-screws at each end. The hull is what is known as a double-ender—that is, has its greatest immersed transverse section at exactly its midlength, and both bodies alike. The *Oxton*, which was put in service in August, 1879, nevertheless, makes her embarkations and discharges of passengers from the side of the vessel, and not from the ends, as in the case of the *Bergen*, and is said to manœuver with great facility.

In England, owing to the absence of ice in the rivers throughout the year, the propulsion of ferry boats by screws offers no difficulty; but with the thick ice which forms on the waters of the Northern United States in winter, there was doubt from this cause—a doubt which has not yet been resolved by the *Bergen*, as there has been no opportunity to use her in ice up to the present.

In December, 1887, the wooden ferry boat *St. Ignace*, 250 feet long, 50 feet extreme breadth and 22 feet depth of hold, was launched at Detroit, Michigan, for service in crossing the Straits of Mackinac. She was fitted with double-screws, one at each end, like the *Bergen*, but, unlike the *Bergen*, these screws were of largely different dimensions, and were driven on separate shafts by different engines. The vessel always proceeded with the large screw aft and pushing, and the small screw forward and pulling, as long as she was not impeded by ice, but when that was the case the small screw was reversed—the large screw continuing to push—so that one screw worked against the other, the propelling thrust on the hull being the difference of the thrusts of the two screws, instead of the sum of the thrusts, as in normal working. The small screw was used in this manner only when the vessel had to be forced through thick ice, the notion

being that the mass of water projected forward and upward by the small screw broke up the ice by lifting it, and then threw it aside so as to allow the vessel to proceed continuously through ice of the maximum thickness. The plan is said to operate successfully.

The *Bergen* was completed in 1889, and was then thoroughly experimented with in a manner which permits the determination of the following facts:

The economic efficiency of the boilers with semi-bituminous coal.

The economic efficiency of the engines in pounds of water, in units of heat, and in pounds of coal, consumed per hour per horse-power developed.

The economic efficiency of the same screw as regards losses of useful effect by slip and by the resistance of their wetted helicoidal surfaces, using, 1st. Only one screw propelling at the stern, the other screw being removed; and, 2d. Using only one screw pulling at the bow, the other screw being removed. In both cases the immersed external surface of the hull was perfectly clean.

The extent to which the resistance of the hull was affected by the operation of the screw at the bow, when the vessel was pulled by that screw alone, the immersed external surface of the hull being perfectly clean.

The resistance of the vessel at the experimental speed in pounds when unaffected by the action of the bow or forward screw, the immersed external surface of the hull being perfectly clean.

The increased resistance of the hull due to the fouling of the painted steel bottom of the vessel, caused by over six months' exposure to the sea water of New York Harbor, the immersed external surface of the hull being cleaned, as far as was accessible, without docking.

The principal lacune of the experiments was in not ascertaining the performance of the vessel with both screws combined, and with a perfectly clean bottom. This would have required another experiment after the docking of the vessel to replace

the screw which had been removed in order to determine the effect of the remaining screw either forward or aft.

The experiments were admirably devised and conducted by Professor J. E. Denton, of "The Stevens Institute of Technology," at Hoboken, New Jersey. They were made with the greatest possible accuracy and completeness.

No pains were spared to ascertain the true speed of the vessel in several ways on the various trials, namely: By long distances between measured points, correcting for velocity of tide by current meters; by alternate and repeated runs over a short measured base, and by the readings of a tested taffrail log towed astern. Indicator diagrams were taken at short intervals from both ends of the three cylinders of the motive triple-expansion engine, and also from the steam cylinders of the several auxiliary pumping engines.

The weight of feed water pumped into the boilers was ascertained from a meter carefully tested under the experimental conditions of pressure, etc. The coal and its refuse were accurately weighed in the dry state. The revolutions of the engine were given by a reliable engine-room counter.

The vessel's draught of water, forward and aft, did not vary sensibly during the trials; the variable weights on board being only the small quantity of coal required for the trials, and the personnel embarked. The trim of the vessel throughout was on even keel when motionless, and, when in motion, the trim remained sensibly the same, even at the maximum experimental speed, as shown by a spirit level.

A sufficient number of zealous and highly intelligent assistants was furnished from the graduating class of "The Stevens Institute," so that the observations have unusual accuracy.

During a part of the experiments, the writer, by the kind invitation of Mr. E. A. Stevens and Professor Denton, was present, and from these gentlemen he has received the observations, including the indicator diagrams, of the experiments, from which data he has analysed the trials and made the calculations and deductions hereinafter found.

Before proceeding to the results of the experiments, it is necessary to give the dimensions of the hull and machinery, which were as follows:

HULL.

The hull is of steel without coppered wooden sheathing, but with smoothly painted bottom, and has the "double-ender" form of immersed solid required for a ferry boat that makes its alternate crossings without turning; that is to say, the two bodies of the hull, on opposite sides of the greatest immersed transverse section, are duplicates, that section being at exactly midlength between the two vertical sternposts, or rather endposts, of the hull. The water-lines of the immersed solid of the hull are extremely sharp below the experimental draught of water, but at and above that draught they are quite full, as required for a ferry boat. The dead rise at the greatest immersed transverse section is very great, giving that section almost a V form.

The upper part of the hull is carried at each end as an overhang 10 feet beyond the endposts, and beneath this overhang are the screw and the rudder, the outmost edge of the latter being in the vertical of the edge of the overhang. From the upper edge of the keel to the bottom of the overhang is 7 feet 7 inches. The lowest point of the periphery of the screw is 4½ inches below the bottom of the keel. A shoe 2 inches deep extends beneath the screw, leaving a clearance between them of 2½ inches. The shoe is extended far enough to receive the lower end of the vertical shaft of the rudder, the axis of which is 6 feet beyond the endpost. From the bottom of the shoe to the bottom of the overhang is 8 feet 9 inches, and the vessel's experimental draught of water being 10 feet from the bottom of the shoe, there were 1 foot 3 inches of the hull above the overhang immersed. Hence, of the depth of the hull from the upper edge of the keel to the water surface, namely 8 feet 10 inches, there were comprised between the endposts 7 feet 7 inches, and between the ends of the upper part of the hull 1 foot 3 inches.

Between the endpost of the hull and the hub of the screw, there are 18 inches in the clear; the hub is 18 inches in length,

and between it and the rudder are 24 inches in the clear. The rudder is 5 feet broad, with vertical edges. It is counterbalanced with the axis 12 inches from the edge adjacent to the endpost of the hull. The weight of the rudder is supported from the deck, the shoe acting only as a steadiment. The rudder is controlled by a powerful steam steering engine.

The deck of the vessel projects beyond the hull 12 feet 6 inches on each side at midlength, and 1 foot at each end, these extreme dimensions being joined by an easy curve, so that the deck has the form of a flat ellipse. The projection is supported at intervals by struts inclined at about 45 degrees from the side of the hull to midway of the projection.

Cabins of light joiner work are erected along each side of the vessel upon the side projections of the deck, for the accommodation of passengers. These cabins are about 13 feet in height and 140 feet in length. They and the space between them are covered by a light wooden deck, on which are erected the two steering houses—one at each end. A very narrow strip at the center of the vessel between the main and upper decks is inclosed to furnish room for the stairs ascending to the upper deck and descending to the fire and engine rooms, for the chimney of the boilers, and for the ventilation of the fire and engine rooms. With the exception of this strip, the space between the cabins is for the transportation of horses, vehicles, etc.

The following are the principal dimensions of the hull:

Extreme length of main deck	200 feet.
Extreme length of hull beneath main deck, also length on water surface	198 feet.
Length of hull from outside to outside of endposts below overhang	178 feet.
Extreme breadth of main deck	62 feet.
Breadth of hull beneath main deck	37 feet.
Breadth of hull at water surface due to experimental draught	32 feet 5 inches.
Breadth of hull at bottom of the overhang, amidship	29 feet 9 inches.
Depth of hull from top of main deck to upper edge of keel.	17 feet 11 inches.
Depth of hull from upper edge of keel to water surface at experimental draught	8 feet 10 inches.
Depth of keel (uniform throughout)	5 inches.

Bottom of shoe below bottom of keel at each end of vessel.	9 inches.
Experimental draught of vessel, at both ends, from bottom of shoe	10 feet.
Experimental draught of vessel, uniform between ends, from bottom of keel	9 feet 3 inches.
Greatest immersed transverse section of the hull	166 square feet.
Water section of the hull	5,292 square feet.
Water section of the hull at the bottom of the overhang	3,890 square feet.
Wetted surface of the keel, rudders, and shoes	648 square feet.
Wetted surface of the body of the hull	5,140 square feet.
Total immersed external, or wetted, surface of the vessel	5,788 square feet.
Displacement of the hull from top of keel to bottom of overhang (35 cubic feet per ton)	398 tons.
Displacement of the hull above bottom of overhang	162 tons.
Total displacement of the hull	560 tons.
Displacement at water section per inch of draught	12.6 tons.
Ratio of length to breadth of hull beneath main deck	5.3513
Ratio of length to breadth of hull on water section	6.1080
Ratio of length between endposts of hull below the overhang to the breadth of the hull at bottom of the overhang.	5.9832
Ratio of the greatest immersed transverse section to its circumscribing parallelogram	0.6752
Ratio of the water section to its circumscribing parallelogram	0.8245
Ratio of the water section at the bottom of the overhang to its circumscribing parallelogram	0.7346
Ratio of the displacement above bottom of overhang to its circumscribing parallelopipedon	0.7067
Ratio of the diplacement below bottom of overhang to its circumscribing parallelopipedon	0.3469
Mean ratio of the total diplacement to its circumscribing parallelopipedon	0.4510
Mean angle of the water lines of the hull	13 degrees 13 minutes.
Angle of dead rise	23 degrees.
Weight of the hull (iron alone, without woodwork)	720,348 pounds.

The above calculations are made by supposing the hull to be composed of two parts, one superimposed upon the other, caused by the overhang at each end necessary to cover the screws and afford sufficient displacement for the weights at the end of the vessel.

ENGINE.

The *Bergen* has one motive engine of the so-called triple expansion type with three cylinders. The engine is vertical, with the axes of the cylinders placed in the vertical central longi-

tudinal plane of the vessel, and immediately over the axis of the main shaft. This shaft extends from the screw at one end of the vessel to the screw at the other end, so that both screws make revolution for revolution. There are three pairs of cranks placed at angles of 120 degrees with each other. The engine is direct acting, the main crossheads working in slipper guides. The small cylinder has a cylindrical slide valve, and the other two cylinders have the usual three-ported slide valve. There are no independent cut-off valves; the main steam valve of each cylinder cutting off the steam by lap. The valves are worked by the usual Stephenson link. With the link in full gear, the valve of the small cylinder cuts off the steam at 0.69 of the stroke of the piston from the commencement, and cushions the back pressure at 0.81 of the stroke. The valves of the two other cylinders cut off the steam at 0.76 of the stroke of the piston from the commencement, and cushion the back pressure at 0.87 of the stroke.

During all the experiments the link was in full gear. The engine was fitted with a small steam cylinder for reversing the link. The engine could be stopped, backed, or started ahead almost instantly and with the slightest effort. The cylinders were not steam jacketed, but were well lagged. Each cylinder was supported on one side by brackets cast on the shell of the condenser, and by a wrought-iron column on the other side. A "separator" was placed in the steam pipe, just in advance of the small cylinder.

There was one surface condenser with horizontal brass tubes arranged with one return; the condensing water was within the tubes and the exhaust steam around them. The condenser was placed alongside the engine and occupied the same length. The entire engine, including condenser, was supported upon a cast-iron bed plate.

There was one air pump, vertical and single acting. It was worked from a beam actuated through links by the crosshead of the large cylinder. The main center of the beam was supported on the shell of the condenser.

The circulating pump, the feed pump, and the bilge pump, were horizontal double-acting duplex pumps. Each was entirely

separate from the motive engine and from each other, and consisted of two pumps worked directly by two horizontal steam cylinders, the valve movement being governed by tappets. The steam was used in the cylinders without expansion.

The following are the principal dimensions of the engine:

Number of cylinders	3.
Diameter of the small cylinder	18½ inches.
Diameter of the piston rod of the small cylinder	4¼ inches.
Net area of the piston of the small cylinder, exclusive of rod	261.710 square inches.
Stroke of the piston of the small cylinder	2 feet.
Space displacement of the piston of the small cylinder, per stroke	3.634861 cubic feet.
Space in clearance and steam passage at one end of small cylinder	0.581578 cubic foot.
Per centum which the space in the clearance and steam passage at one end of the small cylinder, is of the space displacement of the piston of the small cylinder per stroke.	16.
Area of the steam port of the small cylinder	31.4052 square inches
Diameter of the intermediate cylinder	27 inches.
Diameter of the piston rod of the intermediate cylinder	4¼ inches.
Net area of the piston of the intermediate cylinder, exclusive of rod	565.463 square inches.
Stroke of the piston of the intermediate cylinder	2 feet.
Space displacement of the piston of the intermediate cylinder, per stroke	7.853639 cubic feet.
Space in clearance and steam passage at one end of the intermediate cylinder.	0.840339 cubic foot.
Per centum which the space in the clearance and steam passage at one end of the intermediate cylinder, is of the space displacement of the piston of the intermediate cylinder per stroke	10.7
Area of the steam port of the intermediate cylinder	70.683 square inches.
Diameter of the large cylinder	42 inches.
Diameter of the piston rod of the large cylinder	4¼ inches.
Net area of the piston of the large cylinder, exclusive of rod	1,378.347 square inches.
Stroke of the piston of the large cylinder	2 feet.
Space displacement of the piston of the large cylinder per stroke	19.143708 cubic feet.
Space in clearance and steam passage at one end of the large cylinder	2.163239 cubic feet.

Per centum which the space in the clearance and steam passage at one end of the large cylinder, is of the space displacement of the piston of the large cylinder per stroke	11.3
Area of the steam port of the large cylinder	151.628 square inches.
Number of times that the area of the piston of the large cylinder is of the area of the piston of the small cylinder	5.266696
Number of times that the area of the piston of the large cylinder is of the area of the piston of the intermediate cylinder	2.437555
Number of times that the space displacement of the piston of the large cylinder per stroke plus the space at one end of that cylinder in clearance and steam passage, is of the space displacement of the piston of the small cylinder per stroke plus the space at the end of that cylinder in clearance and steam passage	5.053304
Number of air pumps, (vertical and single acting)	1.
Diameter of the air-pump	24 inches.
Diameter of air-pump piston rod	3 inches.
Stroke of the air-pump piston	6 inches.
Space displacement of air-pump piston per stroke, exclusive of rod	1.54625 cubic feet.

CIRCULATING PUMP.

The exhaust steam from the main and auxiliary engines is liquefied in the surface condenser by refrigerating water supplied from outboard by an independent circulating pump consisting of two horizontal double-acting water cylinders operated directly by two horizontal steam cylinders, the valve of one steam cylinder being worked by a tappet from the piston rod of the other, so that the action of the pistons is "at right angles." The steam is used without expansion, and is supplied from the steam chest of the intermediate cylinder of the main engine. The steam cylinders are arranged to exhaust either into the condenser or into the atmosphere.

Number of steam cylinders	2.
Diameter of steam cylinders	12 inches.
Working stroke of the pistons of the steam cylinders	12 inches.
Aggregate area of the steam pistons of the two steam cylinders, excluding rods	221.48 square inches.
Aggregate displacement of the steam pistons of the two steam cylinders, per stroke	1.53806 cubic feet.

Number of water cylinders	2.
Diameter of water cylinders	14 inches.
Working stroke of the pistons of the water cylinders	12 inches.
Aggregate area of the water pistons of the two water cylinders, excluding rods	303.16 square inches.
Aggregate displacement of the water pistons of the two water cylinders, per stroke	2.10528 cubic feet.

The circulating pump lifted its water 4.5 feet.

FEED PUMP.

The boiler is fed with water by an independent steam pump consisting of two horizontal double-acting water cylinders operated directly by two horizontal steam cylinders, the valve of one steam cylinder being worked by a tappet from the piston rod of the other, so that the action of the pistons is "at right angles." The steam is used without expansion, and is supplied from the boilers through a reducing valve which maintains a uniform pressure in the valve chests of the steam cylinders of 30 pounds per square inch above the atmosphere. The steam cylinders are arranged to exhaust either into the condenser or into the atmosphere.

Number of steam cylinders	2.
Diameter of steam cylinders	12 inches.
Stroke of the pistons of the steam cylinders	12 inches.
Aggregate area of the steam pistons of the two steam cylinders, excluding rods	221.48 square inches.
Aggregate displacement of the steam pistons of the two steam cylinders, per stroke	1.53806 cubic feet.
Number of water cylinders	2.
Diameter of water cylinders	4½ inches.
Stroke of the pistons of the water cylinders	12 inches.
Aggregate area of the water pistons of the two water cylinders, excluding rods	27.094 square inches.
Aggregate displacement of the water pistons of the two water cylinders, per stroke	0.376306 cubic foot.

BILGE PUMP.

The bilge pump is the duplicate of the feed pump, except that the water cylinders are 5 inches in diameter instead of 4½ inches.

WEIGHTS.

Weight of the motive engine complete, but exclusive of the screw shaft and the two screws	122,000 pounds.
Weight of the screw shaft	27,000 pounds.
Weight of the two screws	6,000 pounds.
Weight of the auxiliary engines, piping, fresh water tanks filled, etc.	86,000 pounds.
Total weight of machinery, exclusive of boilers and water in them,	241,000 pounds.

BOILERS.

The *Bergen* has two horizontal cylindrical tubular boilers of the locomotive type; that is, the tubes are in direct continuation of the furnace. Each boiler has two cylindrical furnaces. The boilers have no steam superheating surface.

The shell of each boiler is 8 feet in diameter and 23 feet in length. Thickness of the shell, including flat ends, $\frac{1}{2}$ inch. The two boilers are placed side by side, and between them, at top, is a horizontal cylindrical steam drum containing one-third of the steam room. Both boilers discharge their gases of combustion into a sheet-iron breeching at the after end. The chimney, in common for both boilers, rises from the top of this breeching.

The cylindrical furnaces are 3 feet 3 inches in internal diameter, and are made of $\frac{5}{8}$-inch thick steel stiffened by bands. Each furnace has a grate 6 feet 6 inches in length. The bridge walls are of brick, and 8 inches thick, mounted upon cast-iron supports, which form the back of the ashpits. Distance, at center of furnace, between top of bridge water and top of furnace, 1 foot in clear for draught. The entire length of the cylindrical furnace to the combustion chamber is 9 feet 6 inches. At the top of the furnace, at its extreme inner end, is a baffle wall of brick, the bottom of which is horizontal and 9 inches below the top of the furnace. The thickness of this wall is 5 inches, and its purpose is to deflect the gases of combustion downwards, so as to cause them to enter in part the lower tubes. The space for the direct passage of the gases of combustion, between the top of the bridge wall and the bottom of the baffle wall, projected on a vertical plane, is 3 inches high.

Between the furnaces and the tubes is a combustion chamber

2 feet 6 inches long in the clear, and in common for both furnaces. The top of this chamber is horizontal, and its bottom and sides are concentric with the shell.

Between the combustion chamber and the back end of the boiler are, for each boiler, 204 tubes, 2½ inches in outside diameter and 10 feet 11 inches in clear of the tube plates. The back of the combustion chamber and the back end of the boiler, each ½-inch thick, are the tube plates. The tubes are in eleven rows high, with a vertical water space at the center, corresponding with the water space between the furnaces. Thickness of the metal of the tubes one-tenth of an inch.

The boilers were well protected from heat radiation.

The following are the principal dimensions and proportions of each boiler:

Diameter of shell	8 feet.
Length of shell	23 feet.
Number of furnaces	2.
Diameter of furnace	3 feet 3 inches.
Length of grate	6 feet 6 inches.
Grate surface	42.25 square feet.
Length from face of bridge wall to back of combustion chamber	4 feet 3 inches.
Number of tubes	204.
Outside diameter of tubes	2.5 inches.
Inside diameter of tubes	2.3 inches.
Height of chimney above level of grates	53 feet.
Half the cross-section of the chimney	7.0931 square feet.
Cross-area above bridge walls for draught	4.4372 square feet.
Cross-area of the tubes for draught	5.8859 square feet.
Heating surface in the furnaces	192 square feet.
Heating surface in the combustion chamber	82 square feet.
Heating surface in the tubes (inner circumference)	1,341 square feet.
Total heating surface in one boiler	1,615 square feet.
Ratio of heating to grate surface	38.2249
Ratio of grate surface to draught area over bridge walls	9.5218
Ratio of grate surface to draught area through tubes	7.1782
Ratio of grate surface to cross-section of chimney	5.9565
Weight of the two boilers complete, with doors, grates and plates, but without breeching and chimney	100,852 pounds.
Weight of breeching, chimney and boiler covering, etc	5,148 pounds.
Weight of the water in the two boilers	55,000 pounds.
Working pressure in pounds per square inch above atmosphere	160.

SCREWS.

The *Bergen* has two cast-iron screws, duplicates in all respects, one at each end of the vessel, the two being upon the same shaft, operated by the same engine, and making, of course, revolution for revolution. The lowest point of the periphery of the screws is $4\frac{1}{2}$ inches below the bottom of the keel of the vessel. The pitch of the screws is uniform at every point of their blades. The blades taper from the hub to the periphery by straight lines symmetrically on both sides of a perpendicular erected from the center of the hub, when viewed in projection on a plane parallel to the axis. The blades are $3\frac{3}{4}$ inches thick at the fillet of the hub, and $\frac{3}{4}$-inch thick at the periphery.

The following are the dimensions of one screw:

Diameter of the screw	8 feet.
Diameter of the hub	1.3125 feet.
Pitch of the screw (uniform) by measurement of casting	8.915 feet.
Number of blades	4.
Length of the blades at the hub in the direction of the axis	1.5 feet.
Length of the blades at the periphery in the direction of the axis	10 inches.
Fraction used of the pitch at the hub	0.67302
Fraction used of the pitch at the periphery	0.37390
Mean fraction used of the pitch	0.52346
Helicoidal area of the blades	29.71468 square feet.
Projected area of the blades on a plane at right angles to axis	25.60379 square feet.
Projected area of the blades on a plane parallel to axis	15.55555 square feet.
Weight of one screw	3000 pounds.

THE EXPERIMENTS.

The principal object of the experiments was to determine the efficiency of the forward screw as a propeller, both when propelling alone and when propelling in conjunction with the after screw, for the efficiency varies in the two cases, and can vary greatly with the conditions of the trials, as up to the date of these experiments none were known that bore upon these questions. Unfortunately, the fact of the vessel's bottom being foul when the trial with the two screws acting simultaneously

was made, prevents any determination of the efficiency of the forward screw when used in combination with the after one. But, in the case of the trials with the forward screw used alone, and with the aft screw used alone, on the same day and place, the vessel's bottom was newly painted and quite clean. To accomplish the purpose of these trials there had to be ascertained the speed of the vessel in each case, and the horses-power applied to the propulsion of the vessel. This power is very different from the indicated horses-power developed by the engine, because the indicated horses-power includes not only the power applied to the propulsion of the vessel, but the power expended in working the unloaded engine, the power expended in overcoming the friction of the load, the power expended in overcoming the resistance of the water to the surfaces of the screw, and the power expended in the slip of the screw. Of the indicated horses-power, then, the power applied to the propulsion of the vessel is only a fraction, and a very variable one. Hence, to ascertain the power applied to the propulsion, the powers expended in other work must be ascertained and deducted from the indicated horses-power, which is the power given by the indicator diagram.

It is possible to very closely calculate the powers expended in other work than the propulsion of the vessel. For example: experiment has shown about what pressure in pounds per square inch of the steam pistons is required to work unloaded engines of different dimensions of cylinder, and also that this pressure is constant at all speeds of the same unloaded engine. Experiment, too, has shown that, if the power required to work the unloaded engine be deducted from the indicated power, then, of the remainder (called the net horses-power developed by the engine), $7\frac{1}{2}$ per centum will be absorbed by the friction of the load as a constant with the lubrication and journal pressure of ordinary practice. The power expended in overcoming the resistance of the water to both sides of the screw blades can be calculated exactly for any given screw moving in the water with any given speed, provided the resistance of unit of the helicoidal surface moving in its helical direction with unit of speed be known.

Experiment has shown that ordinary metallic surfaces, rolled or cast, moving in water in directions parallel to the surfaces at the speed of 10 feet per second, has the resistance of 0.45 pound per square foot, and that this resistance varies as the square of the speed.

Now, the powers absorbed by the friction of the load and by the resistance of the screw surfaces to the water having been calculated, and their sum subtracted from the net horses-power developed by the engine, the remainder will be the horses-power expended in propelling the vessel, and in the slip of the screw. The slip of the screw being known in fractions of its axial speed (revolutions in a given time multiplied by pitch), the product of the above remainder by this fraction gives the power expended in the slip of the screw, which power being deducted from the above remainder, leaves the horses-power expended in the propulsion of the vessel. This is the method that has been followed in determining the distribution of the indicated horses-power, and there can be no doubt of its substantial accuracy; that is to say, it will give the power applied to the propulsion of the vessel within a very few per centums—quite as accurately, probably, as if a dynamometer had been attached to the screw shaft and this power calculated from the thrust shown by it.

Supposing the preceding method and the dynamometer to be exact, they would give exactly the same result.

The use of a dynamometer in such cases is extremely desirable, but it will rarely be resorted to because of the inconvenience and expense. In its absence, the method proposed is the best that can be done, and has to be adopted if any quantitative information is to be had. A close approximation is certainly better than no knowledge. But the indications of a perfectly made and properly applied dynamometer could not be relied on to exactly show the thrust of the screw upon the hull, for even supposing the thrust of the shaft upon the dynamometer to be exactly shown, there would have still to be added to that thrust an *estimated* quantity for the endwise friction resistance of the shaft equal to the friction due to the weights of the shaft and screw upon the shaft journals. Again, the thrust measured by the dyna-

mometer is the thrust in the direction of the axis of the screw shaft, not the thrust in the horizontal direction, which latter is the thrust required to be known. Now, the axis of the screw shaft is rarely, if ever, horizontal, and for strict exactness the indications of the dynamometer would have to be corrected for the *estimated* obliquity. There is no error resulting from the obliquity of the axis of the screw shaft to the horizon in the case of the calculations of the method, because the slip of the screw, as calculated, is its true slip irrespective of the obliquity of its axis. The greater that obliquity, other things equal, the greater will, indeed, be the slip of the same screw, but the slip as calculated is, nevertheless, the correct slip; that is to say, it is the difference between the space passed over in unit of time by the form of the screw and by the vessel, and this difference represents the loss of useful effect due to the slip of the screw or the recession of the watery fulcrum of the screw under its pressure.

By knowing the number of square feet of immersed external or wetted surface of the vessel, including the surfaces of the keel, rudder, rudder-post, etc., and using the same resistance of 0.45 pounds per square foot of that surface moving with the velocity of 10 feet per second, and varying as the square of the velocity, ascertaining the velocity of the surface from the speed of the vessel corrected for the obliquity the water lines of the hull to the vertical central longitudinal plane of the latter, there can be calculated the horses-power absorbed by the resistance of the wetted surfaces of the hull, which power being deducted from the power applied to the propulsion of the hull as determined by the method, leaves the power overcoming the resistance of the *form* of the hull irrespective of the resistance of its external immersed surface. In this way only can the relative value of form alone, or model pure and simple, be ascertained, the resistance of the hull as a whole being composed of the resistance of its form and the resistance of its wetted surface, the latter being independent of the form in a certain sense, and varying always in the ratio of the square of the velocity, while the resistance of the form varies in that ratio only within certain limits of speed depending on the absolute size of the hull,

above which limits the resistance of the form alone varies in a higher ratio determinable only by experiment. Thus, the relation between the resistance of the form alone of a given vessel, and the resistance of its wetted surface, which relation is constant as long as the speed of the vessel does not exceed the limit where its resistance as a whole remains in the ratio of the square of the speed, changes when the speed of the hull surpasses that limit, the fraction of the power required to overcome the resistance of the hull as a whole becoming less for the wetted surface and more for the form, until, at excessive speeds of vessel, the divergence may be enormous, owing to the fact that the power required to overcome the resistance of the wetted surface always remains in the ratio of the square of the speed, be the form what it may, while the resistance of the form increases in a greater and greater ratio from "squatting" after a certain limit of speed is reached, depending on the dimensions and the shape of the hull, irrespective of the resistance of the wetted surface.

In the case of the *Bergen*, all the experiments were made within the "squatting" limit of speed, that is to say, within the experimental speeds the hull did not sensibly alter trim by rising at the forward end and sinking at the after end, but remained substantially on even keel throughout.

The slip of the same screw was constant during all the experiments, and the net horses-power developed by the engine was in the duplicate ratio of the speed of the vessel.

The experiments were three in number, and will be classed as follows, paying no attention to the order of time:

1st. An experiment made with one screw alone pushing at the after end of the vessel, the other screw having been removed. This trial gives the normal performance of the vessel with one screw aft; and the results are what were obtained from eight single runs in continuous succession over the base of two statute miles in the Hudson river opposite the mouth of the Spuyten Duyvel creek. The indicator diagrams were taken at short intervals, and the times of passing the ends of the base were taken by a stop watch. The revolutions were taken by a reliable engine-room counter. The results of the experiment will be

found in the following Table No. 1. The pressures on the pistons are the mean from all the diagrams taken. The vessel's draught of water (10 feet) is the draught from the bottom of the shoe at the ends where the screw descends below the bottom of the keel. The draught of water of the vessel proper, that is between the shoes, is 9 feet 3 inches from the bottom of the keel.

2d. An experiment made with one screw alone pulling at the forward end of the vessel, the other screw having been removed. This experiment was made immediately after the one above described, on the same day and under the same conditions of method, observations, draught of water, etc., by simply turning the vessel around and running it over the same base the same number of times; the only difference being that the same screw which in the previous experiment pushed the vessel now pulled it. The results of the experiment will be found in the following Table No. 2. Both experiments were made on the 28th of September, 1889.

3d. An experiment made with both screws combined, the forward screw pulling and the after screw pushing. This experiment consisted of a continuous run in the Hudson River from Jersey City to Newburgh and return, occupying exactly nine hours, during which a complete set of indicator diagrams were taken every half hour from each end of each cylinder of the motive engine. Diagrams were also frequently taken from the steam cylinders of the auxiliary engines, namely, those working the feed pump, the bilge pump, and the circulating pump. The feed water consumed was measured by two Worthington water meters, carefully tested for delivery under the experimental conditions of pressure and velocity. One meter measured the water, the steam from which was used in the motive engine alone; the other meter measured the water, the steam from which was used in the auxiliary engines alone. The manner in which this division was accomplished, all the steam being generated in the same boilers, was as follows: During this experiment the exhaust steam from all the auxiliary engines was discharged into the atmosphere, only the exhaust steam from the motive engine being discharged into the surface condenser; consequently, the

quantity of feed water pumped into the boilers was greater than the quantity of water of condensation that entered the hot well, and greater by the quantity of water the steam from which supplied the auxiliary engines. The deficiency of feed water obtained from the hot well under this condition was supplied from the large water tanks kept permanently on board the vessel to make good the losses of feed water by the engines during the regular ferry service. These tanks were kept filled to a mark during the experiment, the water for which purpose passed through a separate meter. The total feed water pumped into the boilers being passed through another meter, the difference of the quantities shown by the two meters was the consumption of feed water by the main engine alone.

By this arrangement, although the weight of feed water pumped into the boilers was obtained separately for the motive engine and for the auxiliary engines, yet the quantity of feed water for the latter purpose was enormously increased over what it would have been had the auxiliary engines exhausted into the surface condenser instead of into the atmosphere. These engines were greatly too large for the work they had to do, and they did it, accordingly, with extravagant consumption of steam, even when their steam was used with condensation. Furthermore, their steam was, in all cases, used without expansion. The division of the feed water allowed the cost of the horse-power per hour in pounds consumed of that water to be exactly and properly determined for the motive engine; and, had the feed and circulating pumps been, like the air-pump, worked by the motive engine, the cost of the horse-power in the case of the auxiliary engines would have been the same as the cost of the horse-power developed by the motive engine without them. Nevertheless, under no circumstances can the cost of the horse-power developed by small auxiliary engines be made as little as the cost of the horse-power developed by large motive engines, hence the employment of auxiliary engines should be avoided as much as possible, if economy of fuel be the object; but, on the other hand, their use is often attended with so much convenience

and advantage that the economy may well be sacrificed to practical requirements.

The coal consumed was semi-bituminous; it and its refuse were weighed carefully in the dry state. It was consumed at the maximum rate of combustion with natural draught; consequently, the power developed by the engine was the maximum that could be sustained permanently under this condition.

The gauges, thermometers and counters employed were quite reliable.

The various steam pressures in the cylinders are the means from all the indicator diagrams taken. These pressures for each cylinder of the motive engine are those at the commencement of the stroke of the pistons, at the point of cutting off the steam, at the end of the stroke of the pistons, against the pistons during their stroke both inclusive and exclusive of the cushioning, at the point where the cushioning commences, all being given in pounds per square inch above zero. There are also given the indicated pressure on the pistons in pounds per square inch, which is the mean pressure above the back pressure including the cushioned pressure; the net pressure on the pistons in pounds per square inch, which is the indicator pressure less the pressure required to work the unloaded engine; the mean pressure of the expanding steam alone in pounds per square inch of the pistons above the zero of pressure. For the small cylinder this expansion pressure is the mean between the point of cutting off the steam and the end of the stroke of the piston; for the other two cylinders it is the mean for the entire stroke above zero, including, of course, the cushioned pressure. The total pressure on the piston of the small cylinder is the indicator pressure plus the back pressure against the piston above zero, exclusive of the cushioning, or for the length of the stroke of the piston between the end of the stroke and the point where the cushioning begins, and reduced in the ratio of that length to the whole length of the stroke. The total pressure so obtained is the proper pressure to use with the entire area of the piston of the small cylinder, and its speed, for calculating the total horses-power developed in the small cylinder due to the weight of feed water pumped

into the boilers, as this method eliminates the effect of the cushioning, the weight of steam cushioned or compressed remaining in the cylinder and not being discharged with the exhaust. If the total horse-power were calculated for the mean pressure of the entire diagram taken down to the zero line, there would be necessary to add the weight of the compressed steam due to the cushioning to the weight of the feed water pumped into the boilers in order to obtain the cost of the total horse-power. As the cushioned steam does not affect the cost of the indicated horse-power and the net horse-power, the pressures producing those powers being above the back pressure including the cushioning, the simpler method seems to be to calculate the total horse-power exclusive of the cushioning, in which case it can be compared, as regards cost, with the same weight of feed water as the indicated horse-power and the net horse-power.

The foregoing method of calculating the total horses-power developed, and the cost thereof in pounds of feed water consumed per hour, shows that the weight of steam imprisoned in the cylinder by the cushioning was exactly equivalent to the power expended in its compression afterwards. The bulk of pressure of the intensity produced by the compression, exactly reproduces the power effecting the compression precisely the same as though an equal bulk of equal pressure were taken from the boiler for that purpose. Hence the economy of the power developed by a cylinder, so far as relates to the equivalency of the weight of steam compressed and the power compressing it, is not affected by cushioning, let the amount of the latter be what it may, and let the cylinder pressures be what they may.

In the case of cylinders having unbalanced slide valves—that is to say, valves free to lift from their seats whenever the pressure beneath them is sufficiently greater than the pressure upon them—the cushioning should not, practically, be carried farther than to produce a cushioned pressure equal to the pressure upon the valve, for if it be carried much farther the valve will be lifted from its seat, and steam will be discharged into the exhaust passage and lost. To carry the compression even as far as this

will, however, be found difficult in most cases, owing to the liquefaction effected of part of the compressed steam, notwithstanding that it has received the heat of compression additionally, by the comparatively cool metallic surfaces of the cylinder with which this steam is in contact. The curve of compression on the indicator diagram will consequently be found to reverse before, and often considerably before, the valve-chest pressure is reached, and this reversal will be found strongly marked in all unjacketed cylinders if the steam valve be set without lead, or rather a little late in opening the steam port. If the valves have the usual steam lead, this reversal is masked by the entering of the valve-chest steam before the piston arrives at the end of its stroke.

The fact of cushioning does not affect the loss by clearance or waste spaces at the cylinder ends, which is precisely the same whether the steam be cushioned or not.

There is, however, a well marked economic gain resulting from cushioning, and proportional to it, as regards the development of steam power in a cylinder, due to, and proportional to, the heat imparted by the compressed steam to the comparatively cool metallic surfaces of the cylinder with which it is in contact. The whole of the heat so imparted is a clear gain, and becomes effective in preventing part of the liquefaction which would otherwise be produced upon the steam at its entrance into the cylinder. Mechanically, cushioning is very desirable as the best and most economic method of bringing the piston and its attachments (piston rod, crosshead, connecting rod, etc.,) slowly and gently to rest at the ends of the stroke, thus avoiding shock and concussion. Steam lead does indeed accomplish the same object, but expensively, as all the steam admitted by it before the piston reaches the end of its stroke merely increases the back pressure against the piston, and to that extent reduces the economy with which the power is developed.

The back pressure steam was sufficiently cushioned in the cylinders of the motive engine of the *Bergen* for all practical purposes.

The indicated and net horses-power developed in the intermediate cylinder are calculated for the respective indicator and net pressures for the *entire* area of the piston, and for the speed of the piston; but the total horses-power therein developed are calculated for only the annular surface which remains of the piston after subtracting from the entire area of the piston the area of the piston of the small cylinder. This subtraction is required because the back pressure against the piston of the small cylinder is included in the back pressure against the piston of the intermediate cylinder, having been taken down to the zero of pressure. The total pressure on the annular space just described of the piston of the intermediate cylinder is the indicated pressure plus the back pressure exclusive of the cushioned steam reduced in the ratio of the fraction of the stroke of the piston before the cushioning commenced to the entire stroke. The pressure of the expanded steam on the annular space just described of the piston of the intermediate cylinder is the indicated pressure plus the back pressure inclusive of the cushioned steam.

The powers developed in the large cylinder have been calculated in a similar manner as for the intermediate cylinder. The indicated and net horses-power were calculated for the respective indicated and net pressures, for the *entire* area of the piston, and for the speed of that piston. The total horses-power therein developed are calculated for only the annular surface which remains of the piston of the large cylinder after subtracting the area of the piston of the intermediate cylinder. The total pressure on this surface is the indicated pressure plus the back pressure exclusive of the cushioned steam pressure reduced in the ratio of the fraction of the stroke of the piston before the cushioning commenced to the entire stroke. The pressure of the expanded steam on the annular space just described of the piston of the large cylinder is the indicated pressure plus the back pressure against the piston inclusive of the cushioned steam.

The aggregate of the indicated horses-power developed in the three cylinders of the motive engine are the indicated horses-power developed by the engine.

The aggregate of the net horses-power developed in the three cylinders of the motive engine are the net horses-power developed by the engine.

The aggregate of the total horses-power developed in the three cylinders of the motive engine are the total horses-power developed by the engine.

The aggregate of the horses-power developed in the three cylinders of the motive engine by the expanding steam alone are the horses-power developed by that steam in the engine.

The total horses-power are the entire power developed by the steam under the experimental conditions, and employed in overcoming all the resistances to the steam pressure, prejudicial and useful, except the compression of the back pressure due to the cushioning. They do not include the latter because the weight of steam imprisoned in the cylinder by the cushioning is not added to the weight of steam drawn from the boiler. When the performances of engines are compared by means of the cost of the total horses-power per hour in pounds of fuel or of steam, or of units of heat consumed, the latter being the proper measure of cost, the effect of the differences of back pressure is eliminated.

The indicated horses-power are the total horses-power less the horses-power required to overcome the back pressure, exclusive of the pressure due to the cushioning. It is the power measured by the indicator diagram.

The net horses-power are the indicated horses-power less the horses-power required to work the unloaded engine. They are the only portion of the total horses-power which is commercially valuable, or which produces mechanical effects external to the engine itself.

The horses-power developed by the expanding steam alone, and the pressure producing them, are only required for the purpose of calculating the weight of steam liquefied in the cylinder to set free the heat transmuted into the power developed by the expanding steam. In calculating this weight of steam the mechanical equivalent of heat is taken at 789¼ foot pounds of work per Fahrenheit unit of heat. After the closing of the cut-off

on the small cylinder all the pressure in that cylinder, and all the pressures in the intermediate and large cylinders, are the pressures of expanding steam. The fact that in these cylinders the steam is cut off does not affect the ratio of the measure of expansion with which the steam is used, that depending wholly on the point of cutting off in the small cylinder, and on the relative displacements of space per stroke of piston by the pistons of the small and large cylinders plus the respective waste spaces at the ends of those cylinders.

In calculating the weight of steam in the cylinder as such at the point of cutting off, the space displacement of the piston in cubic feet up to that point plus the space in cubic feet in the clearance and steam passage at one end of the cylinder are taken as the cubic feet of steam and multiplied by the weight in fractions of a pound per cubic foot of steam of the pressure at the point of cutting off, and from this quantity is deducted the quantity obtained by multiplying the space displacement of the piston in cubic feet during the cushioning plus the space in cubic feet in the clearance and steam passage at one end of the cylinder by the weight in fractions of a pound per cubic foot of steam of the pressure of the back pressure at the point where the cushioning commenced. The pressures, of course, are always above zero.

For the weight of steam present as such at the end of the stroke of the piston, the space displacement in cubic feet of the piston per stroke must be taken plus the space in cubic feet of the clearance and steam passage at one end of the cylinder, and the sum multiplied by the weight in fractions of a pound per cubic foot of the steam of the pressure at the end of the stroke of the piston; then, from this latter quantity must be deducted the weight of the steam in the cushion space in cubic feet plus the waste spaces in cubic feet at one end of the cylinder, obtained by multiplying that sum by the weight of a cubic foot of steam of the pressure at the point where the cushioning began.

The quantities in Table No. 3 are grouped for facility of reference, and are so fully described in the table that further explanation is needless.

Table No. 1.—Containing the data and results of the trial made on the 28th of September, 1889, with the ferry boat *Bergen*, propelled by the aft screw alone—the forward screw having been removed—over a base of 2 statute or 1.73513 geographical miles in the Hudson river opposite the embouchure of the Spuyten Duyvil creek. Light breeze and smooth water. Vessel's mean draught of water, 10 feet. Immersed external surface of hull, clean. Four runs over the base in each direction, making eight single runs.

Steam pressure in boilers in pounds per square inch above atmosphere..	67.3
Vacuum in the condenser in inches of mercury.....	27.0
Number of double strokes made per minute by the steam pistons.....	142.125

INDICATED AND NET PRESSURES ON THE PISTONS OF THE THREE CYLINDERS OF THE MAIN ENGINE.

Indicated pressure on the piston of the small cylinder of the main engine, in pounds per square inch.....	28.500
Net pressure on the piston of the small cylinder of the main engine, in pounds per square inch.....	24.989
Indicated pressure on the piston of the intermediate cylinder of the main engine, in pounds per square inch.....	16.670
Net pressure on the piston of the intermediate cylinder of the main engine, in pounds per square inch.....	15.045
Indicated pressure on the piston of the large cylinder of the main engine, in pounds per square inch.....	6.300
Net pressure on the piston of the large cylinder of the main engine, in pounds per square inch.....	5.633

EQUIVALENT INDICATED AND NET PRESSURES ON THE PISTON OF THE LARGE CYLINDER OF THE MAIN ENGINE.

Indicated pressure in pounds per square inch that would have been upon the piston of the large cylinder had the aggregate indicated horses-power developed by the main engine been developed in that cylinder alone.....	18.551
Net pressure in pounds per square inch that would have been upon the piston of the large cylinder had the aggregate net horses-power developed by the main engine been developed in that cylinder alone...	16.550

HORSES-POWER DEVELOPED BY THE MAIN ENGINE.

Number of indicated horses-power developed by the small cylinder.....	128.493646
Number of net horses-power developed by the small cylinder.....	112.664131
Number of indicated horses-power developed by the intermediate cylinder.....	162.388892

Number of net horses-power developed by the intermediate cylinder....	146.559141
Number of indicated horses-power developed by the large cylinder.....	149.594547
Number of net horses-power developed by the large cylinder.	133.756485
Aggregate number of indicated horses-power developed by the engine..	440.477085
Aggregate number of net horses-power developed by the engine.....	392.979757

SPEED AND SLIP.

Speed of the vessel per hour in geographical miles of 6,086 feet.....	10.492776
Speed of the screw per hour in geographical miles of 6,086 feet.....	12.491400
Slip of the screw per hour in geographical miles of 6,086 feet.....	1.998624
Slip of the screw in per centum of its speed.....	16.000000

DISTRIBUTION OF THE AGGREGATE INDICATED HORSES-POWER DEVELOPED BY THE MAIN ENGINE OF THE *Bergen* DURING THE PRECEDING PERFORMANCE PROPELLING WITH THE AFT SCREW ALONE, THE FORWARD SCREW BEING REMOVED.

	Horses-power.	*Per centum of the net horses-power.*
Aggregate indicated horses-power developed by the main engine.....	440.477085	
Horses-power expended in working the main engine, *per se.*	47.497328	
Aggregate net horses-power applied to the crank pins.....	392.979757	or 100.0000
Horses-power absorbed by the friction of the load.....	29.473482	or 7.5000
Horses-power expended in overcoming the resistance of the water to the surface of the blades of the screw...	46.433102	or 11.8156
Horses-power expended in the slip of the screw.....	50.731708	or 12.9095
Horses-power expended in the propulsion of the vessel.....	266.341465	or 67.7749
Totals.....	392.979757	or 100.0000

THRUST OF THE SCREW.

The horses-power expended in the propulsion of the vessel, according to the above "distribution of the power," being 266.341465, is equal to ($266.341465 \times 33000 =$) 8,789,268.345 foot-pounds of work per minute; and the speed of the vessel being 10.492776 geographical miles per hour, is equal to ($\frac{10.492776 \times 6086}{60} =$) 1,064.317246 feet per minute; hence the resistance of the vessel at that speed, or its equivalent the thrust of the screw, is ($\frac{8789268.345}{1064.317246} =$) 8,257.9404 pounds.

DETERMINATION OF THE HORSES-POWER EXPENDED DURING THE PRECEDING PERFORMANCE WITH THE AFT SCREW ALONE—THE FORWARD SCREW BEING REMOVED—IN OVERCOMING THE RESISTANCE OF THE WATER TO THE IMMERSED EXTERNAL OR WETTED SURFACE OF THE HULL.

Taking the resistance of the water to one square foot of the clean painted surface of the immersed exterior of the vessel moving in it with the velocity of 10 feet per second to be 0.45 pound, and at other velocities to be this quantity modified in the ratio of their squares to the square of 10, and deducing from the speed of the vessel the mean speed of its immersed surface due to the inclination of its horizontal water-lines to its longitudinal central plane, there results for that speed 17.5 feet per second and, consequently, a suface resistance of ($10^2 : 0.45 :: 17.5^2 :$) 1.378125 pounds per square foot moving with that velocity.

As the immersed external or wetted surface of the vessel was 5,788 square feet, the power expended in overcoming its resistance was ($\frac{5788 \times 1.378125 \times 17.5 \times 60}{33000} =$) 253.800511 horses-power; consequently, of the 266.341465 horses-power required to propel the hull alone at the experimental speed, ($\frac{253.800511 \times 100}{266.341465} =$) 95.2914 per centum were expended in overcoming the resistance of the wetted surface to the water, and the remaining 4.7086 per centum were expended in the displacement of water by the immersed solid of the hull, irrespective of the resistance of its immersed surface.

Table No. 2.—Containing the data and results of the trial made on the 28th of September, 1889, with the ferry boat *Bergen* propelled by the forward screw alone—the aft screw having been removed—over a base of 2 statute or 1.73513 geographical miles in the Hudson River opposite the embouchure of the Spuyten Duyvil creek. Light breeze and smooth water. Vessel's mean draught of water, 10 feet above bottom of shoe. Immersed external surface of hull, clean. Four runs over the base in each direction, making eight single runs.

Steam pressure in boilers in pounds per square inch above atmosphere	68.3
Vacuum in the condenser in inches of mercury	27.0
Number of double strokes made per minute by the steam pistons	144.750

INDICATED AND NET PRESSURES ON THE PISTONS OF THE THREE CYLINDERS OF THE MAIN ENGINE.

Indicated pressure on the piston of the small cylinder of the main engine, in pounds per square inch	28.500
Net pressure on the piston of the small cylinder of the main engine, in pounds per square inch	24.989
Indicated pressure on the piston of the intermediate cylinder of the main engine, in pounds per square inch	18.000
Net pressure on the piston of the intermediate cylinder of the main engine, in pounds per square inch	16.375
Indicated pressure on the piston of the large cylinder of the main engine, in pounds per square inch	6.336
Net pressure on the piston of the large cylinder of the main engine, in pounds per square inch	5.669

EQUIVALENT INDICATED AND NET PRESSURES ON THE PISTON OF THE LARGE CYLINDER OF THE MAIN ENGINE.

Indicated pressure in pounds per square inch that would have been upon the piston of the large cylinder had the aggregate indicated horses-power developed by the main engine been developed in that cylinder alone	19.1318
Net pressure in pounds per square inch that would have been upon the piston of the large cylinder had the aggregate net horses-power developed by the main engine been developed in that cylinder alone	17.1314

HORSES-POWER DEVELOPED BY THE MAIN ENGINE.

Number of indicated horses-power developed by the small cylinder	133.483995
Number of net horses-power developed by the small cylinder	114.745012
Number of indicated horses-power developed by the intermediate cylinder	178.583496
Number of net horses-power developed by the intermediate cylinder	162.461375
Number of indicated horses-power developed by the large cylinder	153.228079
Number of net horses-power developed by the large cylinder	137.097535
Aggregate number of indicated horses-power developed by the engine	465.295570
Aggregate number of net horses-power developed by the engine	414.303922

SPEED AND SLIP.

Speed of the vessel per hour in geographical miles of 6,086 feet	9.602340
Speed of the screw per hour in geographical miles of 6,086 feet	12.718545
Slip of the screw per hour in geographical miles of 6,086 feet	3.116205
Slip of the screw in per centum of its speed	24.501269

DISTRIBUTION OF THE AGGREGATE INDICATED HORSES-POWER DEVELOPED BY THE MAIN ENGINE OF THE *Bergen* DURING THE PRECEDING PERFORMANCE PROPELLING WITH THE FORWARD SCREW ALONE, THE AFT SCREW BEING REMOVED.

	Horses-power.	*Per centum of the net horses-power.*
Aggregate indicated horses-power developed by the main engine	465.295570	
Horses-power expended in working the main engine, *per se*,	50 991648	
Aggregate net horses-power applied to the crank-pins	414.303922 or	100.0000
Horses-power absorbed by the friction of the load	31.072794 or	7.5000
Horses-power expended in overcoming the resistance of the water to the surface of the blades of the screw	49.053730 or	11.8400
Horses-power expended in the slip of the screw	81.877703 or	19.7627
Horses power expended in the propulsion of the vessel	252.299695 or	60.8973
Totals	414.303922 or	100.0000

THRUST OF THE SCREW.

The horses-power expended in the propulsion of the vessel, according to the above "distribution of the power," being 252.299695, is equal to ($252.299695 \times 33000 =$) 8,325,889.935 foot-pounds of work done per minute; and the speed of the vessel being 9.602340 geographical miles per hour, is equal to ($\frac{9.602340 \times 6086}{60} =$) 973.997354 feet per minute; hence the resistance of the vessel at that speed, or its equivalent the thrust of the screw, is ($\frac{8325889.935}{973.997354} =$) 8,548.1648 pounds.

Table No. 3.—Containing the data and results of the trial of the ferry boat *Bergen* propelled by the combined forward and aft screws, during a round voyage in the Hudson River on the 18th of September, 1889, between the cities of Jersey City and Newburgh. Light breeze and smooth water. Total distance steamed through the water 112.8872 statute miles of 5,280 feet or 97.9369 geographical miles of 6,086 feet. Vessel's mean draught of water from bottom of shoe, 10 feet. Both screws in use. Bottom of vessel, foul.

ENGINE.

Steam pressure in the boilers, in pounds per square inch above the atmosphere	114.
Position of the throttle-valve	Partly closed.
Vacuum in the condenser, in inches of mercury	25.75
Pressure of the atmosphere, in pounds per square inch	14.70
Pressure in the condenser, in pounds per square inch above zero	2.06
Number of double strokes made per minute by the steam pistons	142.125
Fraction completed of the stroke of the pistons of all the cylinders when the steam was released	0.96
Fraction completed of the stroke of the piston of the small cylinder when the steam was cushioned	0.81
Fraction completed of the stroke of the pistons of the other two cylinders when the steam was cushioned	0.87
Fraction completed of the stroke of the piston of the small cylinder when the steam was cut off	0.69
Fraction completed of the stroke of the pistons of the other two cylinders when the steam was cut off	0.76
Number of times the steam was expanded in the small cylinder	1.364706
Number of times the steam was expanded in the engine	6.896274
Number of Fahrenheit degrees imparted in the condenser to the refrigerating water	31.5

FEED WATER.

Temperature, in degrees Fahrenheit, of the feed water in the hot well	118.
Number of pounds of feed water pumped into the boilers per hour to supply the steam used in the main engine	12,063.
Number of pounds of feed water pumped into the boilers per hour to supply the steam used in the auxiliary engines	2,348.
Total number of pounds of feed water pumped into the boilers per hour	14,411.
Number of Fahrenheit units of heat imparted per hour in the boilers to the feed water supplying the main engine with steam	13,286,746.47564
Number of Fahrenheit units of heat imparted per hour in the boilers to the feed water supplying the auxiliary engines with steam	2,586,195.86546
Total number of Fahrenheit units of heat imparted per hour to the total feed water pumped into the boilers	15,872,942.34110

TEMPERATURES.

Temperature, in degrees Fahrenheit, of the steam in the boilers	346.35
Temperature, in degrees Fahrenheit, of the gases of combustion leaving the boilers	750.

COAL.

Number of pounds of semi-bituminous coal consumed per hour to supply the main engine with steam	1,339.310250
Number of pounds of semi-bituminous coal consumed per hour to supply the auxiliary engines with steam	260.689750
Number of pounds of combustible consumed per hour to supply the main engine with steam	1,171.896469
Number of pounds of combustible consumed per hour to supply the auxiliary engines with steam	228.103531
Total number of pounds of semi-bituminous coal consumed per hour	1,600
Total number of pounds of ash, clinker and unconsumed fine pieces of coal withdrawn per hour from ashpits and furnaces	200
Total number of pounds of combustible or of semi-bituminous coal less the refuse, consumed per hour	1,400
Per centum of coal in ash, clinker and unconsumed fine pieces	12.5
Pounds of coal consumed per hour per square foot of grate	18.9349
Pounds of combustible consumed per hour per square foot of grate	16.5480
Fraction of a pound of coal consumed per hour per square foot of heating surface	0.4954
Fraction of a pound of combustible consumed per hour per square foot of heating surface	0.4334

STEAM PRESSURES IN SMALL CYLINDER PER INDICATOR.

Pressure on the piston of the small cylinder at the commencement of its stroke, in pounds per square inch above zero	112.25
Pressure on the piston of the small cylinder at the point of cutting off the steam, in pounds per square inch above zero	93.00
Pressure on the piston of the small cylinder at the end of its stroke, in pounds per square inch above zero	67.00
Back pressure, including cushioning, against the piston of the small cylinder, in pounds per square inch above zero	53.75
Back pressure, excluding cushioning, against the piston of the small cylinder, in pounds per square inch above zero	50.70
Back pressure against the piston of the small cylinder at the point where the cushioning begins, in pounds per square inch above zero	52.20
Indicated pressure on the piston of the small cylinder, in pounds per square inch	44.325
Net pressure on the piston of the small cylinder, in pounds per square inch	40.814
Total pressure on the piston of the small cylinder, in pounds per square inch above zero	85.392
Pressure on the piston of the small cylinder of the expanding steam alone, in pounds per square inch above zero	77.70

STEAM PRESSURE IN INTERMEDIATE CYLINDER PER INDICATOR.

Pressure on the piston of the intermediate cylinder at the commencement of its stroke, in pounds per square inch above zero,	50.20
Pressure on the piston of the intermediate cylinder at the point of cutting off the steam, in pounds per square inch above zero,	42.70
Pressure on the piston of the intermediate cylinder at the end of its stroke, in pounds per square inch above zero	29.70
Back pressure, including cushioning, against the piston of the intermediate cylinder, in pounds per square inch above zero	19.50
Back pressure, excluding cushioning, against the piston of the intermediate cylinder, in pounds per square inch above zero	18.60
Back pressure against the piston of the intermediate cylinder at the point where the cushioning begins, in pounds per square inch above zero	16.50
Indicator pressure on the piston of the intermediate cylinder, in pounds per square inch	24.700
Net pressure on the piston of the intermediate cylinder, in pounds per square inch	23.075
Total pressure on the piston of the intermediate cylinder, in pounds per square inch above zero	40.882
Pressure on the piston of the intermediate cylinder of the expanding steam alone, in pounds per square inch above zero	44.40

STEAM PRESSURES IN LARGE CYLINDER PER INDICATOR.

Pressure on the piston of the large cylinder at the commencement of its stroke, in pounds per square inch above zero	16.50
Pressure on the piston of the large cylinder at the point of cutting off the steam, in pounds per square inch above zero	12.50
Pressure on the piston of the large cylinder at the end of its stroke, in pounds per square inch above zero	10.00
Back pressure, including cushioning, against the piston of the large cylinder, in pounds per square inch above zero	4.00
Back pressure, excluding cushioning, against the piston of the large cylinder, in pounds per square inch above zero	3.50
Back pressure against the piston of the large cylinder at the point where the cushioning begins, in pounds per square inch above zero	3.25
Indicator pressure on the piston of the large cylinder, in pounds per square inch	9.503
Net pressure on the piston of the large cylinder, in pounds per square inch	8.836
Total pressure on the piston of the large cylinder, in pounds per square inch above zero	12.548
Pressure on the piston of the large cylinder of the expanding steam alone, in pounds per square inch above zero	13.503

POUNDS OF STEAM ACCOUNTED FOR PER HOUR BY INDICATOR IN THE CYLINDERS OF THE MAIN ENGINE.

Number of pounds of feed water pumped into the boilers per hour to supply the steam used in the main engine	12,063.000000
Number of pounds of steam present per hour in the small cylinder at the point of cutting off the steam, calculated from the pressure there	9,990.411250
Number of pounds of steam present per hour in the small cylinder at the end of the stroke of its piston, calculated from the pressure there	10,094.185028
Number of pounds of steam present per hour in the intermediate cylinder at the end of the stroke of its piston, calculated from the pressure there	9,738.497109
Number of pounds of steam present per hour in the large cylinder at the end of the stroke of its piston, calculated from the pressure there	8,823.211594
Number of pounds of steam liquefied per hour in the small cylinder to furnish the heat transmuted into the power developed by the expanding steam alone in that cylinder	294.341932
Number of pounds of steam liquefied per hour in the intermediate cylinder to furnish the heat transmuted into the power developed by the expanding steam alone in that cylinder	632.261475
Number of pounds of steam liquefied per hour in the large cylinder to furnish the heat transmuted into the power developed by the expanding steam alone in that cylinder	471.642817
Aggregate number of pounds of steam liquefied per hour in the three cylinders of the main engine to furnish the heat transmuted into the power developed by the expanding steam alone in that engine	1,398.246224

DIFFERENCE BETWEEN THE WEIGHT OF STEAM VAPORISED IN THE BOILERS AND THE WEIGHT ACCOUNTED FOR BY THE INDICATOR IN THE CYLINDERS OF THE MAIN ENGINES.

Difference, in pounds per hour, between the weight of steam present in the small cylinder of the main engine at the point of cutting off the steam, and the weight of steam evaporated in the boilers to supply this engine with steam	2,072.588750
Difference, in per centum of the weight of steam evaporated in the boilers to supply the main engine with steam, between that weight and the weight of steam present in the small cylinder of the main engine at the point of cutting off	17.181371
Difference, in pounds per hour, between the weight of steam present in the small cylinder of the main engine at the end of the stroke of its piston, and the weight of steam evaporated in the boilers to supply this engine with steam	1,968.814972

Difference, in per centnm of the weight of steam evaporated in the boilers to supply the main engine with steam, between that weight and the weight of steam present in the small cylinder of the main engine at the end of the stroke of its piston.....	16.321106
Difference, in pounds per hour, between the weight of steam present in the intermediate cylinder of the main engine at the end of the stroke of its piston, and the weight of steam evaporated in the boilers to supply this engine with steam.....	2,324.502891
Difference, in per centum of the weight of steam evaporated in the boilers to supply the main engine with steam, between that weight and the weight of steam present in the intermediate cylinder of the main engine at the end of the stroke of its piston,	19.269691
Difference, in pounds per hour, between the weight of steam present in the large cylinder of the main engine at the end of the stroke of its piston, and the weight of steam evaporated in the boilers to supply this engine with steam.....	3,239.788406
Difference, in per centum of the weight of steam evaporated in the boilers to supply the main engine with steam, between that weight and the weight of steam present in the large cylinder of the main engine at the end of the stroke of its piston..	26.857236

HORSES-POWER DEVELOPED BY MAIN ENGINE.

Number of indicated horses-power developed by the small cylinder,	199.841459
Number of net horses-power developed by the small cylinder.....	184.011941
Number of total horses-power developed by the small cylinder...	384.994063
Number of horses-power developed by the expanding steam alone (calculated above zero) in the small cylinder.....	105.094290
Number of indicated horses-power developed by the intermediate cylinder	240.612217
Number of net horses-power developed by the intermediate cylinder	224.782466
Number of total horses-power developed by the intermediate cylinder	213.928792
Number of horses-power developed by the expanding steam alone (calculated above zero) in the intermediate cylinder.....	231.291341
Number of indicated horses-power developed by the large cylinder,	225.650252
Number of net horses-power developed by the large cylinder.....	209.812231
Number of total horses-power developed by the large cylinder...	175.719361
Number of horses-power developed by the expanding steam alone (calculated above zero) in the large cylinder.....	189.092965
Aggregate number of indicated horses-power developed by the engine.....	666.103928
Aggregate number of net horses-power developed by the engine..	618.606638
Aggregate number of total horses-power developed by the engine..	774.642216
Aggregate number of horses-power developed by the expanding steam alone (calculated above zero) in the engine.....	525.478596

FEED-PUMP PERFORMANCE PER INDICATOR.

Number of double strokes made per minute by the steam pistons of the feed-pump	54½
Steam pressure on the steam pistons of the feed-pump from the beginning to the end of their stroke, in pounds per square inch above zero	41.0
Mean back pressure against the steam pistons of the feed-pump from the beginning to the end of their stroke, in pounds per square inch above zero	23.5
Back pressure against the steam pistons of the feed-pump at the beginning of their stroke, in pounds per square inch above zero,	20.0
Indicator pressure on the pistons of the feed-pump, in pounds per square inch	17.5
Net pressure on the pistons of the feed-pump, in pounds per square inch	16.5
Total pressure on the pistons of the feed-pump, in pounds per square inch above zero	41.0
Indicated horses-power developed by the steam pistons of the feed-pump	12.723915
Net horses-power developed by the steam pistons of the feed-pump	11.996834
Total horses-power developed by the steam pistons of the feed-pump	29.810315
Number of pounds weight of steam present per hour in the steam cylinders of the feed-pump at the end of the stroke of their pistons, calculated from the pressure there, assuming the waste spaces at the ends of the cylinders to be one-sixteenth of the space displacement of their pistons per stroke. No release and no cushioning	1,332.899844
Number of cubic feet of space displaced per hour by the water pistons of the feed-pump	1,222.975
Number of cubic feet of feed-water pumped per hour by the feed-pump	233.517
Number of horses-power theoretically required to supply the boilers with water	2.185719

NOTE.—The feed-pump, when drawing water from the hot-well, draws in, simultaneously, considerable air, owing to the variation of the water level in the hot-well To separate such air from the water for the purpose of preventing it from entering the boilers, a pipe was led from the base of the air chamber of the pump to the top of the hot-well, whereby a large portion of the displacement of the pump's water pistons simply circulated between the pump and the hot-well. Supposing the water cylinders of the pump to fill, normally, to three-fourths of their capacity, then, during the experiment, the pump was operated at about four times the speed necessary for supplying the boilers with water. During the experiment the steam cylinder exhausted into the atmosphere.

BILGE-PUMP PERFORMANCE PER INDICATOR.

Number of double strokes made per minute by the steam pistons of the bilge-pump	48.833333

Steam pressure on the steam pistons of the bilge-pump from the beginning to the end of their stroke, in pounds per square inch above zero	22.0
Mean back pressure against the steam pistons of the bilge-pump from the beginning to the end of their stroke, in pounds per square inch above zero	18.0
Back pressure against the steam pistons of the bilge-pump at the beginning of their stroke, in pounds per square inch above zero,	18.0
Indicator pressure on the pistons of the bilge-pump, in pounds per square inch	4.0
Net pressure on the pistons of the bilge-pump, in pounds per square inch	3.0
Total pressure on the pistons of the bilge-pump, in pounds per square inch above zero	22.0
Indicated horses-power developed by the steam pistons of the bilge-pump	2.621968
Net horses-power developed by the steam pistons of the bilge-pump	1.966476
Total horses-power developed by the steam pistons of the bilge-pump	14.420822
Number of pounds weight of steam present per hour in the steam cylinders of the bilge-pump at the end of the stroke of their pistons, calculated from the pressure there, assuming the waste spaces at the ends of the cylinders to be one-sixteenth of the space displacement of their pistons per stroke. No release and no cushioning	560.700389

CIRCULATING PUMP PERFORMANCE PER INDICATOR.

Number of double strokes made per minute by the steam pistons of the circulating pumps	22.316667
Steam pressure on the steam pistons of the circulating pump from the beginning to the end of their stroke, in pounds per square inch above zero	28.5
Mean back pressure against the steam pistons of the circulating pump from the beginning to the end of their stroke, in pounds per square inch above zero	16.0
Back pressure against the steam pistons of the circulating pump at the beginning of their stroke, in pounds per square inch above zero	16.0
Indicator pressure on the pistons of the circulating pump, in pounds per square inch	12.5
Net pressure on the pistons of the circulating pump, in pounds per square inch	11.5
Total pressure on the pistons of the circulating pump, in pounds per square inch above zero	28.5
Indicated horses-power developed by the steam pistons of the circulating pump	3.744466

Net horses-power developed by the steam pistons of the circulating pump	3.444909
Total horses-power developed by the steam pistons of the circulating pump	8.537383
Number of pounds weight of steam present per hour in the steam cylinders of the circulating pump at the end of the stroke of their pistons, calculated from the pressure there, assuming the waste spaces at the ends of the cylinders to be one-sixteenth of the space displacement of their pistons per stroke. No release and no cushioning	374.942385

ECONOMIC RESULTS FOR THE MAIN ENGINE.

Pounds of feed water consumed per hour per indicated horse-power developed by the main engine	18.109787
Pounds of feed water consumed per hour per net horse-power developed by the main engine	19.500276
Pounds of feed water consumed per hour per total horse-power developed by the main engine	15.572350
Number of Fahrenheit units of heat consumed per hour per indicated horse-power developed by the main engine	19,946.957099
Number of Fahrenheit units of heat consumed per hour per net horse-power developed by the main engine	21,478.506145
Number of Fahrenheit units of heat consumed per hour per total horse-power developed by the main engine	17,152.107387
Number of pounds of semi-bituminous coal consumed per hour per indicated horse-power developed by the main engine	2.010663
Number of pounds of semi-bituminous coal consumed per hour per net horse-power developed by the main engine	2.165043
Number of pounds of semi-bituminous coal consumed per hour per total horse-power developed by the main engine	1.728940
Number of pounds of combustible consumed per hour per indicated horse-power developed by the main engine	1.759330
Number of pounds of combustible consumed per hour per net horse-power developed by the main engine	1.894413
Number of pounds of combustible consumed per hour per total horse-power developed by the main engine	1.512823

HORSES-POWER DEVELOPED BY THE AUXILIARY ENGINES.

Indicated horses-power developed by the feed-pump engine	12.723915
Indicated horses-power developed by the bilge-pump engine	2.621968
Indicated horses-power developed by the circulating-pump engine	3.744466
Net horses-power developed by the feed-pump engine	11.996834
Net horses-power developed by the bilge-pump engine	1.966476
Net horses-power developed by the circulating-pump engine	3.444909
Total horses-power developed by the feed-pump engine	29.810315
Total horses-power developed by the bilge-pump engine	14.420822

Total horses-power developed by the circulating-pump engine....	8.537383
Aggregate indicated horses-power developed by the above auxiliary engines.....	19.090349
Aggregate net horses-power developed by the above auxiliary engines.....	17.408219
Aggregate total horses-power developed by the above auxiliary engines.....	52.768520

ECONOMIC RESULTS FOR THE AUXILIARY ENGINES.

Pounds of feed water consumed per hour per indicated horse-power developed by the auxiliary engines.....	122.940846
Pounds of feed water consumed per hour per net horse-power developed by the auxiliary engines.....	134.878818
Pounds of feed water consumed per hour per total horse-power developed by the auxiliary engines.....	44.496226
Number of Fahrenheit units of heat consumed per hour per indicated horse-power developed by the auxiliary engines.....	135,471.376948
Number of Fahrenheit units of heat consumed per hour per net horse-power developed by the auxiliary engines.....	148,561.772198
Number of Fahrenheit units of heat consumed per hour per total horse-power developed by the auxiliary engines.....	49,010.202777
Number of pounds of semi-bituminous coal consumed per hour per indicated horse-power developed by the auxiliary engines..	13.655578
Number of pounds of semi-bituminous coal consumed per hour per net horse-power developed by the auxiliary engines.....	14.975096
Number of pounds of semi-bituminous coal consumed per hour per total horse-power developed by the auxiliary engines.....	4.940251
Number of pounds of combustible consumed per hour per indicated horse-power developed by the auxiliary engines.....	11.948631
Number of pounds of combustible consumed per hour per net horse-power developed by the auxiliary engines.....	13.103209
Number of pounds of combustible consumed per hour per total horse-power developed by the auxiliary engines.....	4.322720

DIFFERENCE BETWEEN THE WEIGHT OF STEAM VAPORIZED IN THE BOILERS AND THE WEIGHT ACCOUNTED FOR BY THE INDICATOR IN THE AUXILIARY ENGINES.

Number of pounds of feed water pumped into the boilers per hour to supply the steam used in the auxiliary engines.....	2,348.000000
Aggregate number of pounds of steam present per hour in the steam cylinders of the auxiliary engines at the end of the stroke of their pistons, calculated from the pressure there per indicator.....	2,268.542618
Difference, in pounds per hour, between the weight of steam present in the steam cylinders of the auxiliary engines at the end of the stroke of their pistons and the weight of feed water vaporised in the boilers to supply these engines with steam.....	79.457382

Difference, in per centum of the weight of feed water vaporised in the boilers to supply the auxiliary engines with steam, between that weight and the weight of steam present in the steam cylinders of these engines at the end of the stroke of their pistons	3.384045

EQUIVALENT STEAM PRESSURES ON THE PISTON OF THE LARGE CYLINDER.

Indicated pressure, in pounds per square inch, that would have been upon the piston of the large cylinder, had the aggregate indicated horses-power developed by the engine been developed in that cylinder alone	28.0522
Net pressure, in pounds per square inch, that would have been upon the piston of the large cylinder, had the aggregate net horses-power developed by the engine been developed in that cylinder alone	26.0519
Back pressure, exclusive of cushioning, in pounds per square inch above zero, that would have been against the piston of the large cylinder, had the equivalent back pressures against the pistons of the small and intermediate cylinders been added to the back pressure against the piston of the large cylinder	5.5677
Total pressure, in pounds per square inch above zero, that would have been upon the piston of the large cylinder, had the aggregate total horses-power developed by the engine been developed in that cylinder alone	32.6232
Per centum which the indicated pressure is of the total pressure	85.9885
Per centum which the net pressure is of the total pressure	79.8570
Per centum which the back pressure, exclusive of cushioning, is of the total pressure	16.1048

SPEED AND SLIP.

Speed of the vessel per hour in geographical miles of 6,086 feet	10.881883
Speed of the screws per hour in geographical miles of 6,086 feet	12.491400
Slip of the screws per hour in geographical miles of 6,086 feet	1.609517
Slip of the screws in per centum of their speed	12.885

VAPORISATION.

Number of pounds of feed water that would have been vaporised per hour, had it been supplied at the temperature of 100 degrees Fahrenheit and vaporised under the atmospheric pressure,	14,717.158278
Number of pounds of feet water that would have been vaporised per hour, had it been supplied at the temperature of 212 degrees Fahrenheit and vaporised under the atmospheric pressure,	16,436.721788
Pounds of water vaporised from 100 degrees by one pound of coal	9.198224
Pounds of water vaporised from 100 degrees by one pound of combustible	10.512256

Pounds of water vaporised from 212 degrees by one pound of coal	10.272951
Pounds of water vaporised from 212 degrees by one pound of combustible	11.740516
Pounds of water vaporised per hour per square foot of heating surface under the experimental conditions	4.461600

NOTE.—The above vaporisation is too small for the conditions stated, because the temperature of the water and steam in the boilers was that which was due to the pressure there, namely, 346.35 degrees Fahrenheit instead of 212 degrees, as it would have been had the vaporisation been effected under the atmospheric pressure. Now the utmost fraction of the total heat of combustion which can be utilized is what remains of the temperature of the gases of combustion after deducting the temperature of the steam in the boiler, for, evidently, this temperature is the lowest at which the gases can leave the boiler. Hence, other things remaining the same, had the vaporisation been effected under the atmospheric pressure, it would have been as much greater than the vaporisation given in Table No. 3, as was due to the difference between the steam temperatures 212 and 346.35 degrees. Assuming the temperature of the gases of combustion at the moment of their generation to be 1800 degrees Fahrenheit, the difference between the experimental temperature of the steam and 212 degrees being 134.35 degrees, the tabular vaporisations are too small by $(\frac{134.35 \times 100}{1800} =)$ 7.464 per centum; consequently, they should be increased in the ratio of (100.000—7.464=) 92.536 to 100.000 for strict accuracy under the assumed conditions, which would make the vaporisation by one pound of what remains of the semi-bituminous coal after deducting the furnace refuse (11.740526÷0.92536=), 12.687512 pounds of water from the temperature of 212 degrees Fahrenheit and under the atmospheric pressure

DISTRIBUTION OF THE AGGREGATE INDICATED HORSES-POWER DEVELOPED BY THE MAIN ENGINE OF THE *Bergen* DURING THE PRECEDING PERFORMANCE WITH TWO SCREWS.

	Horses-power.	*Per centum of the net horses-power.*
Aggregate indicated horses-power developed by the main engine	666.103928	
Horses-power expended in working the main engine *per se*,	47.497290	
Aggregate net horses-power applied to the crank-pins	618.606638 or	100.0000
Horses-power absorbed by the friction of the load	46.395498 or	7.5000
Horses-power expended in overcoming the resistance of the water to the surface of the blades of the two screws	92.866205 or	15.0122
Horses-power expended in the slip of the two screws	61.763595 or	9.9843
Horses-power expended in the propulsion of the vessel	417.581340 or	67.5035
Totals	618.606638 or	100.0000

THRUST OF THE SCREWS.

The thrust of the screws, as it would have been measured by a dynamometer directly applied to the screw shaft during the foregoing performance, calculated from the data given therein, and in the above distribution of the power, is as follows:

The horses-power expended in the propulsion of the vessel, according to the distribution of the power, being 417.581340, is equal to (417.581340 × 33,000=) 13,780,184.22 foot-pounds of work per minute; and the speed of the vessel being 10.881883 geographical miles per hour, is equal to ($\frac{10.881883 \times 6086}{60}$=) 1103.7856656 feet per minute; hence the resistance of the vessel at that speed, or its equivalent the thrust of the two screws, is ($\frac{13780184.22}{1103.7856656}$=) 12,484.4747 pounds.

RESULTS.

The most important and interesting portion of the experiments is what relates to the performance of the forward screw relatively to that of the after screw, and, in making the comparison, the performance of the latter screw when propelling the vessel alone must be taken as the standard.

By reference to tables 1 and 2, the after screw when employed alone has the slip of 16.000000 per centum, while the forward screw when employed alone has the slip of 24.501269 per centum, or about one-half more. Now, as exactly the same screw was employed in both cases, this great difference of slip could be the result either of insufficient water reaching the forward screw, or of increased resistance of the vessel due to the action of the forward screw.

The latter fact is easily determined by comparing the speed of the vessel in the two cases with the thrusts of the screws producing it. With the aft screw propelling alone, the vessel made the speed of 10.492776 geographical miles per hour, the thrust of the screw being 8,257.9404 pounds. With the forward screw propelling alone, the vessel made the speed of 9.602340 geographical miles per hour, the thrust of the screw being 8,548.1648 pounds. Now, the thrusts of the screws should be in the ratio of the squares of the speeds of the vessel, if the resistance of the latter was the same in the two cases at equal speed. The squares of the two speeds compare as 1.194061 to 1.000000, and the two respective thrusts of the screw compare as 1.000000 to 1.035145. Hence, the resistance of the vessel when propelled at equal speeds

by both screws will compare inversely as ($\frac{1.194061}{1.000000}$=) 1.194061 for the aft screw, and as ($\frac{1.000000}{1.035145}$=) 0.966048 for the forward screw; or, the resistance of the vessel at equal speed in the case of the two screws will be ($\frac{1.194061}{0.966048}$=) 1.236026 for the forward screw, and 1.000000 for the aft screw. The action of the forward screw, therefore, has increased the resistance of the vessel 23.6026 per centum.

The accuracy of the above determination can be tested by comparing the equivalent net pressures on the piston of the large cylinder with the respective thrusts of the screw, as in the case of the same engine working the same screw, the quotient of the division of the thrust of the screw by the net pressure will be a constant, let the slip of the screw or the resistance of the vessel be what it may. When the propulsion was done by the aft screw alone, this constant was ($\frac{8257.9404}{16.5500}$=) 498.97; and when the propulsion was done by the forward screw alone, this constant was ($\frac{8548.1648}{17.1314}$=) 498.97—exactly the same.

When the same vessel, under different conditions giving different resistances, is propelled by the same screw, the slips of the screw in geographical miles per hour for the same speed of vessel will be, in the different cases, in the ratio of the square roots of the resistance of the vessel. In the case of the *Bergen*, propelled by the aft screw alone at the experimental speed of 10.492776 geographical miles per hour, the velocity of the slip of the screw was 1.998624 geographical miles per hour, the slip being 16 per centum. Now, when the vessel was propelled by the forward screw alone, its resistance was 1.236026 relatively to 1.000000 when it was propelled by the aft screw alone. Hence, at the same speed of vessel (10.492776 miles per hour) in both cases, the velocity of the slip of the forward screw should be ($\sqrt{}$1.000000 : 1.998624 : $\sqrt{}$1.236026 :) 2.222070 geographical miles per hour, which would make the slip of the forward screw in per centum of its speed ($\frac{2.222070 \times 100}{10.492776+2.222070}$=) 17.476186. The experimental slip was 24.501269 per centum, corresponding for the speed of vessel, 10.492776 miles per hour, to the velocity of slip, 3.405174 geographical miles per hour. Had, therefore,

the action of the forward screw been as efficient as the action of the after screw, the slip of the forward screw would have been only 17.476186 per centum instead of its experimental slip of 24.501269 per centum; the difference being due to the reaction of part of its surface against the end of the vessel instead of against free water. At the same speed of vessel (10.492776 miles per hour), had the efficiency of the forward screw been equal to that of the after screw, the velocity of the slip of the forward screw would have been only 2.222070 miles per hour, whereas the velocity of the experimental slip was 3.405174 miles per hour; and, as the efficiency is inversely as the squares of these velocities, the two cases compare inversely as 2.222070^2 and 3.405174^2, or as 1.000000 and 2.348352; that is to say, the propelling efficiency of the forward screw was only $(\frac{1.000000}{2.348352} =)$ 0.425830 of that of the after screw, the remaining 0.574170 of its efficiency having been expended in reaction against the end of the hull.

The great excess of the slip of the forward screw over what is due to the normal resistance of the vessel is thus accounted for, as instead of the whole propelling action of that screw being against free water, only 0.425830 of that action was against free water. The excessive slip of the forward screw was due to the reaction of a large part of its propelling effect against the end of the vessel, and to the increased resistance of the vessel caused thereby, this increased resistance resulting from the shock of the water of the slip projected against the end of the vessel, and from the increased speed of this water over the wetted surface of the hull, increasing, correspondingly, the resistance of that surface. Whatever, therefore, would tend to decrease the slip of the forward screw would, other things equal, tend to decrease the additional resistance of the hull due to the slip, as well as the useless expenditure of power in that slip itself. Evidently, the sharper the front end of the vessel, the less would be the reaction of the screw against it; and, indeed, if the end of the vessel were made sufficiently blunt or full, the entire propelling effect of the forward screw would be lost, as none of its reaction would then be against free water. A decrease in the slip of the forward screw could only be made, other things remaining the

same, by increasing the quantity of its surface acting upon the water in a given time, and this involves a proportionally useless expenditure of power in overcoming the resistance of the wetted surface of the blades of the screw. Only the difference between the gain and the loss could be utilized. The precise quantity of additional acting surface that would give the highest economic result would have to be experimentally found. The *Bergen* is so excessively sharp below water, that nothing more can be practically accomplished in that direction. Probably the use of twin screws at each end of the vessel would enable the forward ones to obtain a greater proportional reaction against free water, as they would not be opposite the center of the end of the vessel, but winged out; and they would increase the vessel's facility of manœuvering, which would be an important consideration, especially for a ferry boat. Doubling the number of motive engines, however, would require a double number of engineers to manage them, and the money loss from this must be deducted from the gain. There are also other expenses which are increased by doubling the number of engines, the aggregate power remaining constant.

The equivalent net pressures on the piston of the large cylinder in the above two cases should be in the ratio of the squares of the speeds of the vessel multiplied by the respective resistances of the vessel. The speed of the vessel when propelled by the aft screw alone was 10.492776 geographical miles per hour, and, when propelled by the forward screw alone, 9.602340 miles per hour, the squares of which speeds compare as 1.194061 and 1.000000. The respective resistances of the vessel compare as 1.000000 and 1.236026. Consequently, the equivalent net pressure should compare as $(\frac{1.236026}{1.194061} =)$ 1.035145 for the case of the forward screw, and 1.000000 for the case of the aft screw. The experimental equivalent net pressures in pounds per square inch of the piston of the large cylinder were, in the two cases, 16.5500 for that of the aft screw, and 17.1314 for that of the forward screw, which numbers are in the ratio of 1.035130, or exactly the same.

The comparisons hereinbefore made for the resistances of the hull when propelled by the aft screw alone, and when propelled by the forward screw alone, do not answer the commercial question of the losses of useful effect in the two cases. For that purpose the quotients of the cubes of the speeds of the vessel divided by the net horses-power developed by the motive engine in the two cases must be compared as follows: The speeds compare as 1.092731 and 1.000000, the cubes of which numbers are 1.304787 and 1.000000. The net horses-power developed by the motive engine compare respectively as 1.000000 and 1.054263; hence, the economic results compare as $(\frac{1.304787}{1.000000} =)$ 1.304787 and $(\frac{1.000000}{1.054263} =)$ 0.947581; consequently, propelling by the aft screw alone was $(\frac{1.304787 - 0.947581 \times 100}{0.947581} =)$ 37.6966 per centum more economical than propelling by the forward screw alone; or, propelling by the forward screw alone was $(\frac{1.304787 - 0.947581 \times 100}{1.304787} =)$ 27.3766 per centum less economical than propelling by the aft screw alone.

Heretofore there have been compared only the two cases of propelling by the aft screw alone and by the forward screw alone. Unfortunately, no comparison can be made of these with the case of propelling by the combined after and forward screws, owing to the foulness of the bottom when this trial (Table No. 3) was made. In point of time the trial with the combined screws was made first and without cleaning the vessel's bottom beyond the parts accessible without docking, although she had not been docked and her bottom cleaned for more than six months, this period extending over midsummer, when the marine accretions are most rapid and the largest. As the resistance of the *Bergen* was almost entirely that of her wetted surface, this considerable amount of foulness influenced it almost pro rata. The experiment with the combined screws was made on the 18th of September, 1889, after which the vessel was docked, the bottom cleaned, and one screw removed. The trials with the aft screw alone (Table No. 1), and with the forward screw alone (Table No. 2), were then made on the 28th of September, 1889.

The results of the trial of the combined screws are only available for the determination, with a close degree of approximation, of the increase in the resistance of the vessel caused by the foulness of the bottom—a most important fact to know—and for the determination of the economic performance of the engines and boilers under the experimental conditions.

First, as regards the relative resistances of the hull when clean and when foul. This is obtained inversely, as before, by dividing the squares of the speeds of the vessel by the respective thrusts of the screws producing them. The after screw propelling the vessel alone gave it a speed of 10.492776 geographical miles per hour by a thrust of 8,257.9404 pounds. The combined screws gave the vessel the speed of 10.881883 geographical miles per hour by a thrust of 12,484.4747 pounds. These speeds compare as 1.000000 to 1.037083, the squares of which are 1.000000 and 1.075541. The thrusts of the screws compare as 1.000000 to 1.511814, consequently the resistance of the foul hull was, inversely, as $(\frac{1.075541}{1.511814}=)$ 0.711424 to $(\frac{1.000000}{1.000000}=)$ 1.000000 for the clean hull, or the resistance of the foul hull was $(\frac{1.000000-0.711424\times100}{0.711424}=)$ 40.5631 per centum greater than that of the clean hull. From this, however, must be deducted the increased resistance of the hull when the forward screw is used. When that screw was used alone, its slip was 24.501269 per centum and the resulting increase of resistance was 23.6026 per centum. But, in the case of the combined screws, the slip of the forward screw is only 12.885 per centum or about one-half; consequently, as the effect of the slip of the forward screw against the vessel should be as the square of the velocity of the slip, the increased resistance will be something less than one-fourth of what it was when the forward screw was used alone, or, say, 5.5631 per centum, leaving the resistance due to the fouling alone (40.5631—5.5631=) 35 per centum of what the resistance would be without the fouling. To overcome this resistance there will be required for equal speeds of vessel, a consumption of practically at least one-third more fuel, which is a very serious objection to a painted steel hull in sea water, and will warrant a considerable increase in the first cost of the vessel

to provide the wooden casing sheathed with copper that only can prevent it.

The steam cylinders of the motive engine were not steam-jacketed, and the waste spaces at their ends were enormous; consequently a very serious loss of economic effect from both causes was sustained. Nevertheless, the indicated horse-power was obtained for the consumption per hour of 18.109787 pounds weight of steam, or, more accurately, of 19,946.957099 Fahrenheit units of heat. The liquefaction of steam in the cylinders due to all causes, was from 17 to 27 per centum, going from the small to the large cylinder, about one-half of which could have been saved by complete steam-jacketing, reducing the consumption of steam, say, one-tenth, making it for the indicated horse-power 16.3 pounds weight per hour, or 17,952 Fahrenheit units of heat. If, from this, there be made some deduction on account of the excessive waste spaces at the ends of the cylinders, the cost of the indicated horse-power need not to have exceeded, say, 16 pounds weight of steam per hour, or 17,628 Fahrenheit units of heat, which, considering the small dimensions of the cylinders, would be a satisfactory economic performance.

With the helm put "hard over," and the screws kept at about the same number of revolutions per minute, the vessel described a circle when propelled by the forward screw alone about double the diameter of the circle described when propelled by the aft screw alone, the turning being made in the same direction in both cases.

There was no difficulty in steering when the vessel was propelled by the forward screw alone, and very little helm was required. Of course, the steering was done in all cases by the aft rudder alone, the forward rudder being locked in the direction of the keel.

From each end of the vessel, and in its central longitudinal plane, there was protruded a stiff wooden beam extending exactly horizontal by spirit level about fourteen feet beyond the edge of the deck when the vessel was at rest. A pulley was inserted at the outer end of each beam. Over each pulley a line was passed having a heavy weight at its outer end, its

inner end being brought inboard, the line lying along the upper side of the beam. The inner end rested upon a scale of inches. Now, when the vessel was at rest, the bottom of the weight attached to the outer end of the line was adjusted to just touch the water, the position of the inner end upon the scale being noted, and, when the screw was making 143½ revolutions in the two cases of propelling by the forward screw alone, and propelling by the aft screw alone, the bottom of the weight was again adjusted, and the difference in the position of the inner end of the line upon the scale of inches noted. The following were the results: 1st. When the vessel was propelled by the forward screw alone, the weight had to be lifted 5½ inches at the bow and 13 inches at the stern. 2d. When the vessel was propelled by the aft screw alone, the weight had to be lifted 6½ inches at the bow and 5 inches at the stern. 3d. The less height (8 inches) which the weight had to be lifted at the stern when the vessel was propelled by the aft screw alone than when it was propelled by the forward screw alone, was doubtless due to the depleting action of the aft screw on the column of water rising vertically to fill the cavity at the stern left by the advance of the vessel. 4th. Similar measurements similarly made simultaneously at the opposite sides of the vessel, about midway its length, showed no variation of the weight from its position when the vessel was at rest.

Fig I.

Fig II.

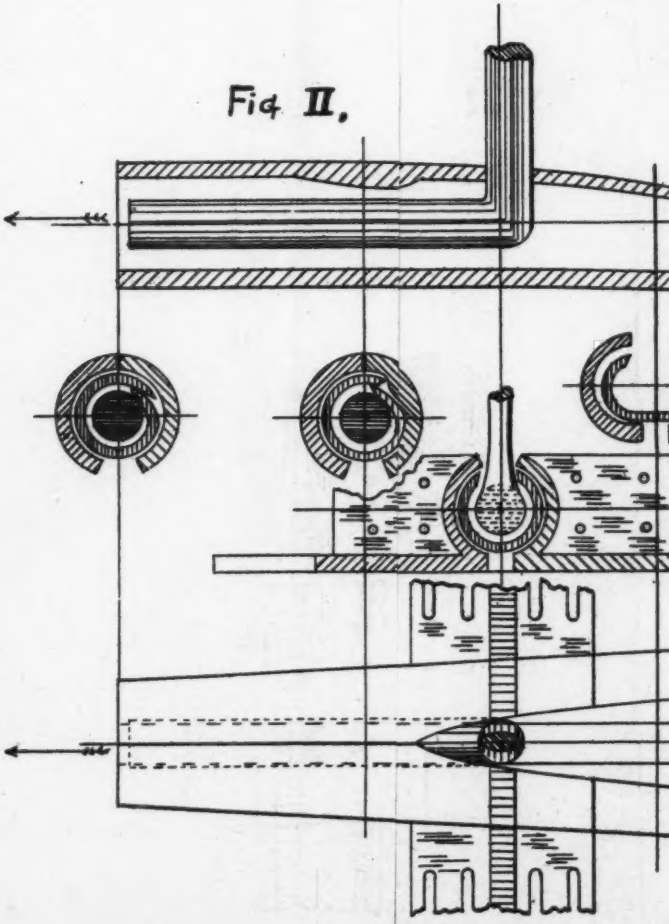
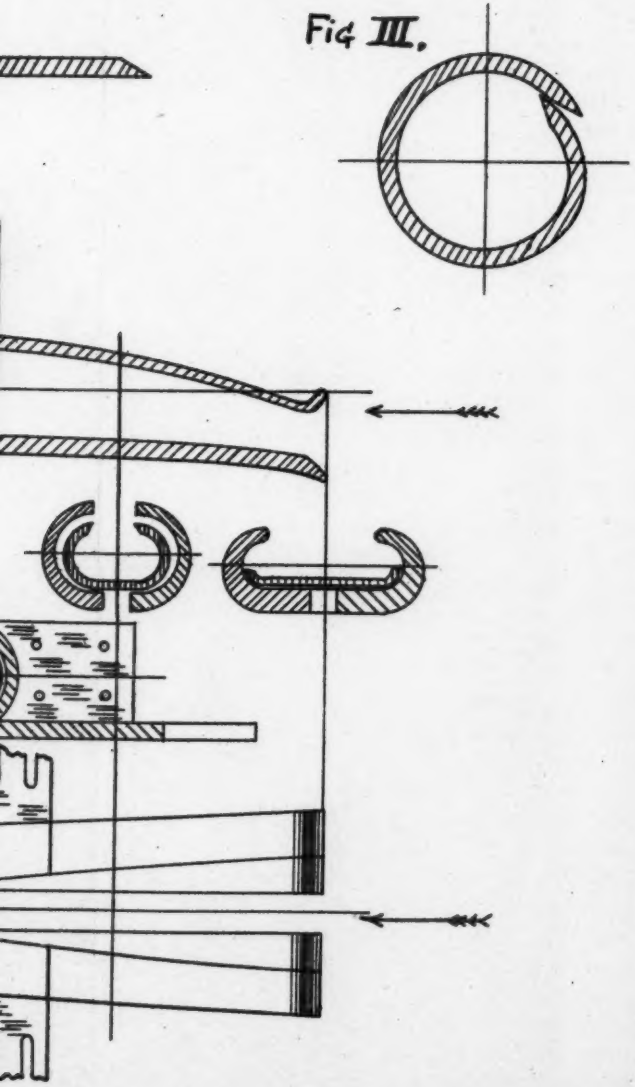


Fig III.



XV.

THE MANUFACTURE AND INSPECTION OF IRON AND STEEL BOILER TUBES.

By Passed Assistant Engineer Reynold T. Hall, U. S. Navy.

[Read before the Philadelphia Branch of the Society; Chief Engineer L. W. Robinson, U. S. Navy, in the Chair.]

In this paper a general description only is intended of the process of manufacturing lap-welded boiler tubes of the kind used in the Service.

The stock used, low-carbon steel or iron, is first rolled to the required width, and at the same time scarfed on the edges by the rolls or in a special planing machine. These sheets, commonly called "skelp," are about six inches longer than the tube to be made therefrom and have a width equal to about the circumference of tube plus three-fourths of an inch for tubes of two to three inches diameter. Fig. I shows a section of rolled skelp.

The skelp is next heated and drawn through the bending die (fig. II), which gives it a cylindrical form and laps the edges, as shown in fig. III. The skelp is drawn through the die by tongs attached to a "buggy" moved by an endless chain. This process is closely watched, and any irregularities in distance between surfaces are remedied by the hand-hammer while the skelp is still hot. The skelp is then put in the welding furnace, and, when at a welding heat, is pushed through the opposite side of furnace from which it entered and is started through the rolls. Fig. IV shows the elevation, and fig. VII the plan and location of the three sets of rolls through which the tube is passed. A, figs. IV, V, and VII, shows the several views of the first set of rolls. The space between the rolls through which the tube passes is not round, but slightly larger in diameter horizontally than vertically. This prevents a ridge being

formed on either side of the tube by the metal squeezing between the extreme edges of the rolls, which would be the case were the space perfectly round.

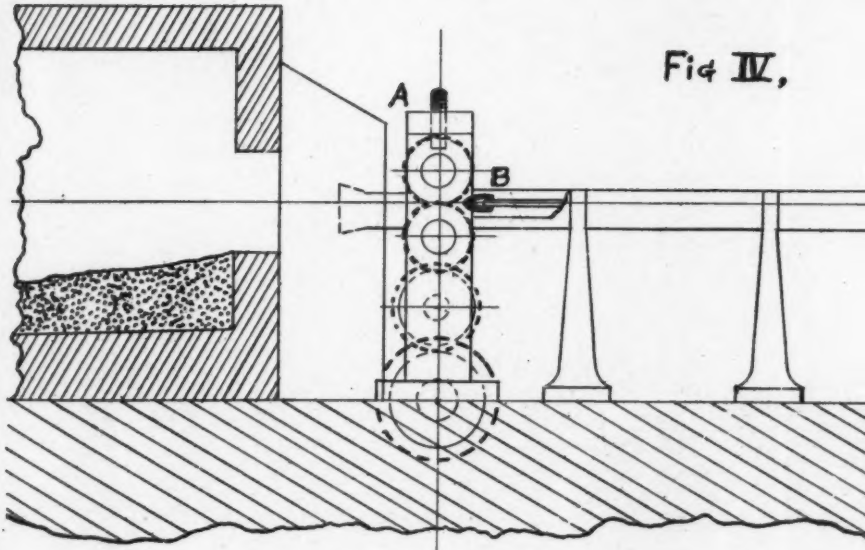
The anvil or mandrel upon which the weld is made is a "ball" or cone of cast iron of diameter suitable to the tube. B, figs. IV and VII, shows the "ball" in position, and fig. IX its sections. A round bar lying in the triangular trough, of which fig. VIII is a section, holds the "ball" in position, and is itself secured as shown at C, figs. IV and VII. The tube as it passes the rolls lies around the bar, which is withdrawn by means of the "bar-puller" (D, figs. IV and VII), the lever and stop being raised. The cone is removed with tongs. The "bar-puller" consists of two grooved wheels, which are revolved in a horizontal plane, and, by means of universal couplings (D, fig. IV) and levers, are pressed against the bar to remove it.

The bar being removed, the tube is rolled across the table (shown by arrow, fig. VII), and while there is examined to see if the weld is complete. If incomplete, the tube is returned to the furnace and then re-rolled in the same direction as before (*i. e.*, front end first). If the weld is complete, the tube is passed through the finishing rolls, where its cylindrical form is perfected. E, fig. VII, shows plan of finishing rolls.

Thence the tube is passed from the triangular trough over the table F, fig. VII (as shown by arrow), to the trough leading to the straightening rolls. Fig. VI shows the end elevation, and G, fig. VII, the plan of these rolls. Their surfaces are hyperbolic longitudinally and round transversely; their axes are inclined and they revolve in opposite directions. They are driven from a shaft through universal couplings. The action of these rolls in giving the tube a spiral motion removes all irregularities such as twist, warp, bend and buckle. The tube finally arrives at the cooling table (H, fig. VII) and is examined for imperfections.

The speed of the skelp through the welding rolls is from 500 to 600 feet per minute; through the finishing rolls from 400 to 500 feet; and somewhat less through the straightening rolls.

Fig IV.



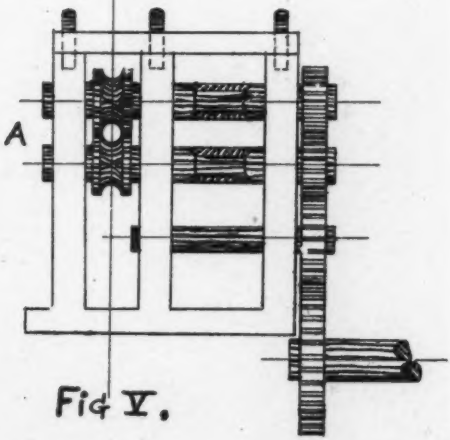
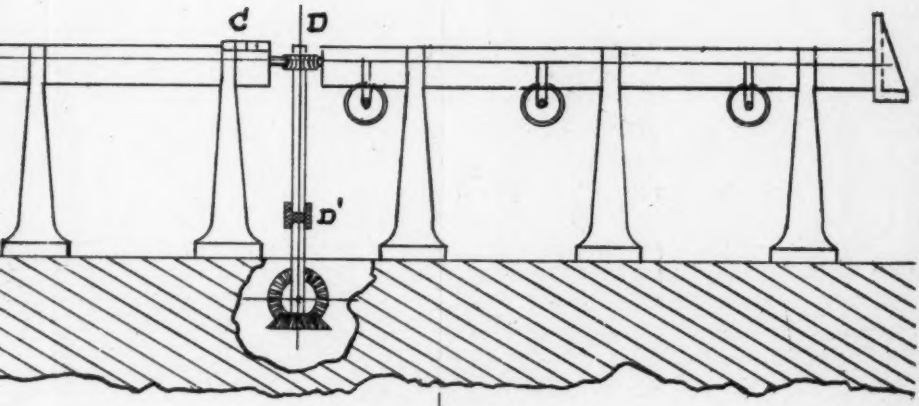


Fig V.

IV,



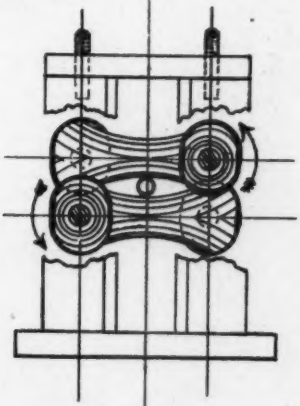


Fig VI,

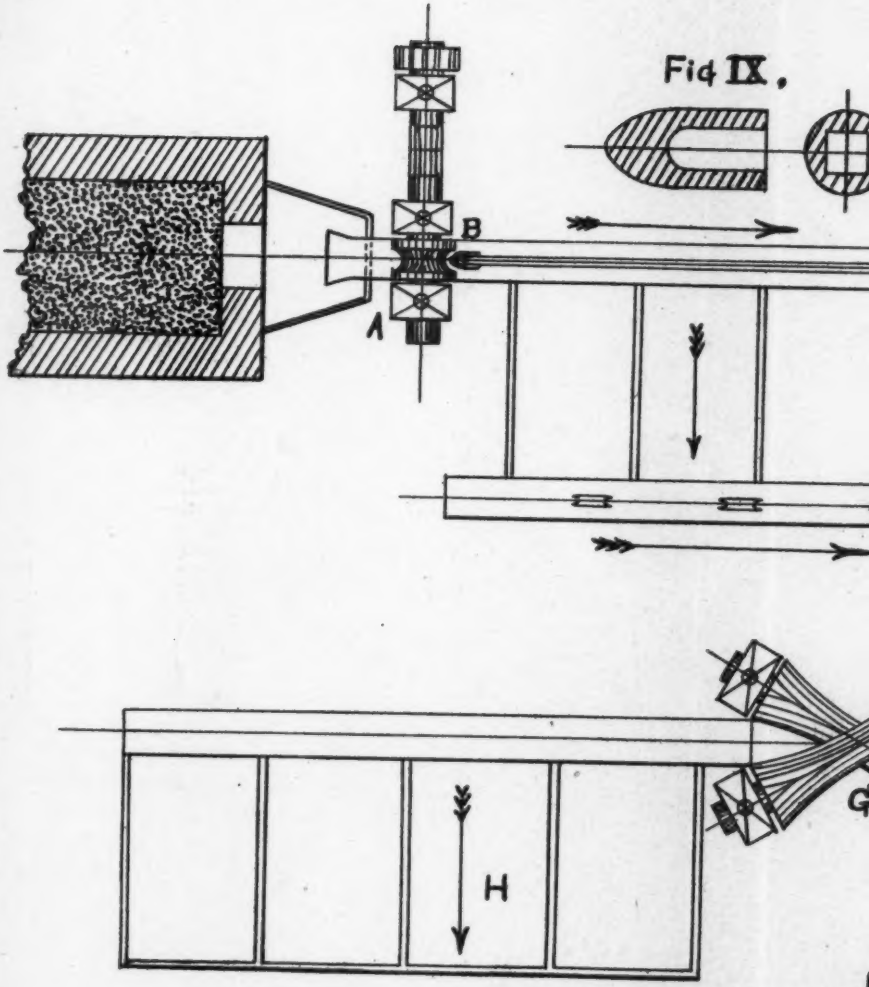


Fig IX.
B
A
H
G

IX.

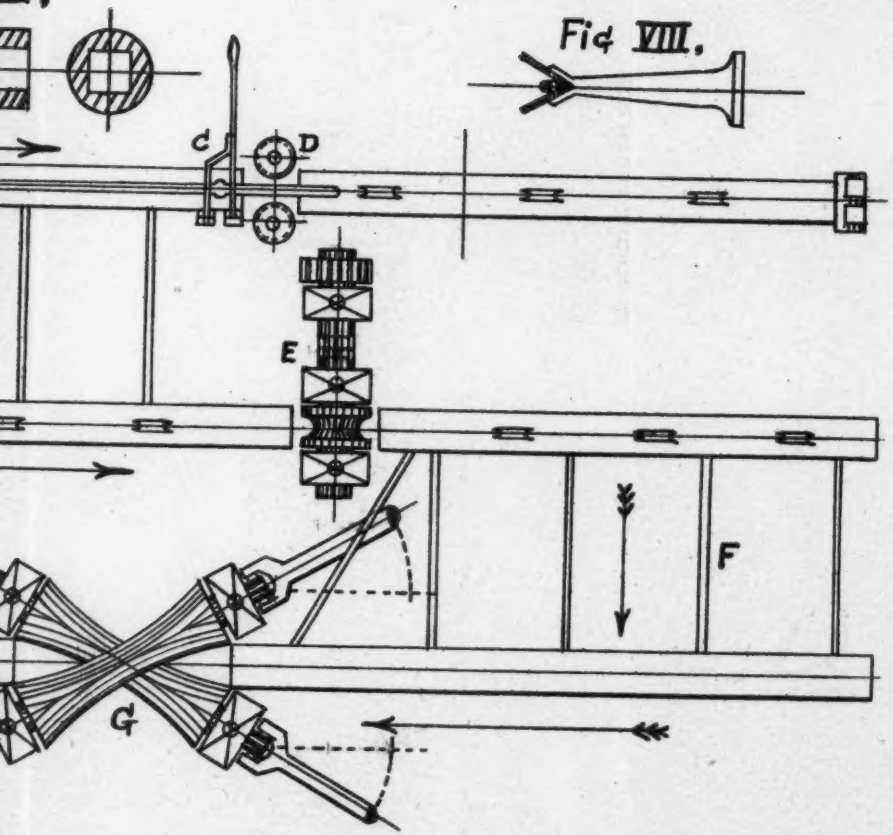


Fig VII.

Fig VIII.

Fig X.

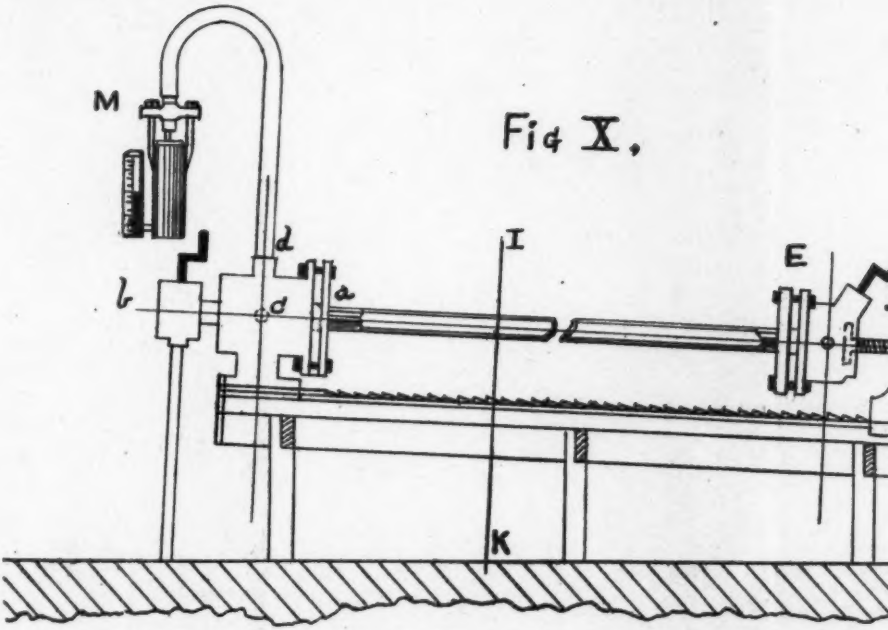


Fig XII.

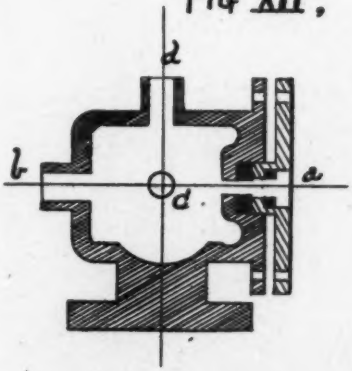
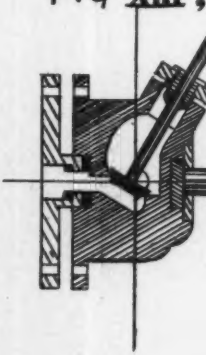


Fig XIII.



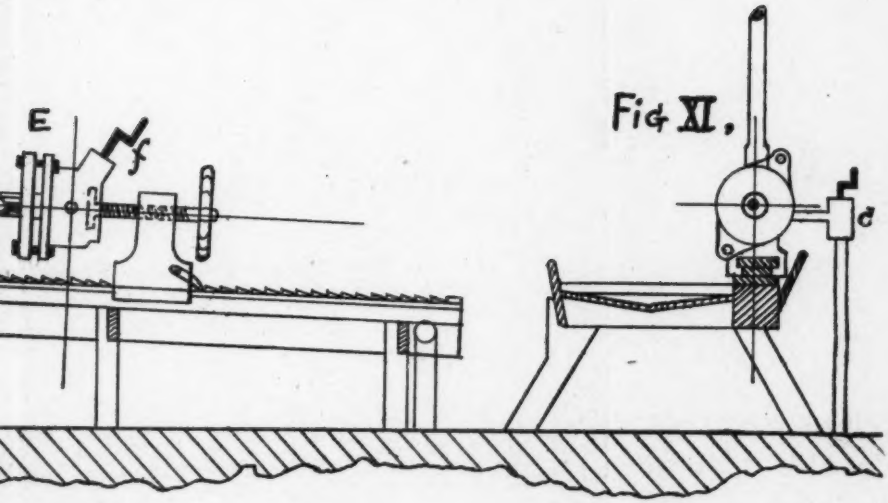


Fig XI.

ig XIII.

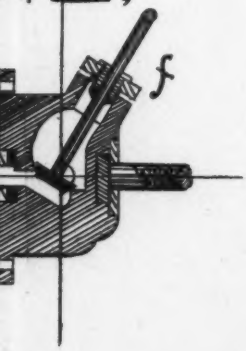
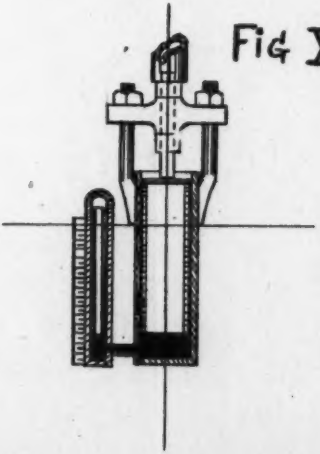


Fig XIV.



Should any perceptible bend remain in the tube, it is straightened by hand between two rollers, one above the other, after which it is taken to the pipe lathe and cut to the proper length.

The finished tube is finally tested to the desired hydrostatic pressure on a testing table. This apparatus consists of a trough and table about 25 feet long, whose elevation is shown in fig. X, and its section (through I-K fig. X) in fig. XI. Fig. XII gives an enlarged view of the "fixed head," and fig. XIII of the "movable head." At *a* (fig. XII), is shown the mode of attaching the face plates for any desired size of pipe. Against the hemp packing in these detachable face plates the tube to be tested is secured. The tube is filled through *b*, with water at ordinary hydrant pressure, a non-return valve being fitted in *b*. The pressure from the pump is admitted at *c* (figs. X, XI and XII). At *d* is a stand pipe connecting with the mercurial gauge (M, fig. X) of which fig. XIV is a section. The movable head (fig. XIII) has face plates similar to those of the fixed head, and, by means of the rack and pawl, can be set for any length of tube. The valve *f* is simply a drain valve to relieve the tube of pressure before its removal after testing.

The ends of the tubes are swelled by driving a pin of the required diameter into the end of the tube the necessary distance while the tube is hot, and, after removing the pin, hammering the end between the dies of a trip hammer to the proper external diameter. The tube ends for about three inches in length are annealed by cooling in a mixture of lime and charcoal (proportion, two-thirds lime to one-third charcoal). About twenty hours are required for cooling in this mixture.

Stay tubes (two to two-and-a-half inches diameter) are made as above described but of heavier material, according to the thickness of finished tubes; thus the rough thickness of material for number 6, B. W. G. tubes is $\frac{5}{8}$ inch, and for number 9 B. W. G. tubes is $\frac{5}{16}$ inch. The tubes are rolled, as already described, to $3\frac{1}{4}$ inches outside diameter, and then successively through 3-inch, $2\frac{3}{4}$-inch and $2\frac{1}{2}$-inch rolls to finished size. This process, it is claimed, gives a more homogeneous metal in the finished tube.

When the ends of the tubes are made heavier than the body (reinforced,) it is done either by upsetting the ends of the tube for the distance required, or by welding to the ends collars of sufficient length and thickness (called "safe-ends"). The former process is an unsatisfactory one for tubes of $2\frac{1}{2}$ inches diameter, on account of the labor and time necessary to complete them, the work being done by hand. The latter process is oftener employed because it is cheaper, more satisfactory, and, by using a steam hammer and dies, more expeditious and exact. This is particularly true with steel, where the number of times the metal can be beneficially heated is limited.

It may be of interest to add a few words in regard to the quality of the material used. To insure a perfect weld, the metal must be soft and ductile. Generally, the better qualities of iron only are used, but in special cases, as of one-inch tubes, where there is a great excess of strength, an inferior grade of iron is used for the body of the tube, and "safe-ends" of best charcoal iron are welded on to stand the expanding into the tube sheets.

Steel for tubes is comparatively free from sulphur and phosphorus, the redshort and coldshort elements, and to secure a ductile metal easily welded the carbon is low. In six samples of low-carbon steel (from tubes inspected by the writer), analyses showed: Carbon, .07 to .09 per cent.; phosphorus .066 to .068 per cent.; sulphur, .03 to .038 per cent. The maximum allowable percentages of these elements vary with different makers. The process of manufacture is so trying, however, that only good material can stand it.

The preliminary inspection of raw material at the tube works consists of a chemical analysis for carbon and of the bending tests usually given to sheet metal of the same kind. A sample of the metal is rolled into a tube, and, when cold, portions of this tube are crushed flat, bent back on themselves, and flanged at the ends. If these tests are all passed satisfactorily, the metal is fit for the best tubes. Failure in some of the tests will not, however, prevent its use for inferior qualities.

The tests for boiler tubes for our new ships prescribed by the Bureau of Steam Engineering are considered the severest, both

in number and character, that have been imposed, and nothing but the best material and workmanship will enable tubes to pass them.

These tests are as follows:—

Every tube must be tested by hydrostatic pressure to at least five hundred pounds per square inch and carefully inspected, while under pressure, for any signs of defects or weakness. From the waste ends of one in every ten stay tubes and one in every twenty common tubes test pieces will be taken which must successfully pass the following tests:

1. A strip cut from each piece, after being heated to a cherry red and quenched in water at about 80° Fahrenheit, must stand being bent flat back on itself without showing cracks or flaws.

1*a*. After annealing, the ends must stand flanging to a half-inch collar when cold without crack.

2. After annealing, a portion of the tube one inch long must stand crushing (in the direction of its axis) under a steam hammer till flattened upon itself without cracks or flaws.

3. If the tubes are lap welded, a piece of sufficient length will be heated to a cherry red and, while at this heat, a taper-pin (also heated) will be driven in till the tube splits. Not more than half the test pieces must part in the weld.

4. A strip about one inch wide and four inches long, cut lengthwise of the tube, must stand bending, while at a dull red heat, around a half-inch bar without signs of distress.

5. All tubes will be carefully examined for surface defects and laminations, and none accepted that are not entirely free from them; and any tubes which may show such defects afterwards, while being worked or pickled, will be rejected and others must be furnished by the makers.

6. If safe-ends are welded on, instead of the tube ends being swelled or upset, any tube which shows defective welds while the threads are being cut will be rejected, and another must be furnished by the makers.

These requirements will repay careful study. It will be seen that they provide almost perfectly for determining that the tubes can stand every form of stress that will come upon them both

during their incorporation into the boiler and in the work of steaming. Tests 1, 1*a* and 2 are designed to show the ductility and capability of withstanding expanding, bending over, &c. Tests 3 and 4 show the soundness of the weld and capacity to withstand increased temperatures. Tests 1*a*, 2 and 3 are very severe, and only the best material will endure them.

During an inspection of several thousand boiler tubes of low-carbon steel for use in our new vessels the writer observed that the percentages of failures to the total number inspected were about as follows: Under the hydrostatic test and inspection for visible defects of whatever kind, about ten per cent. failed; but these were almost entirely on account of surface and other defects, such as blisters, pits, laminations, warping, cracks, uneven lap, imperfect lap, or irregular thickness of tube. A tube that would not stand 500 pounds hydrostatic pressure was very rare, even though it showed some of the defects above enumerated.

All the tubes passed tests, 1, 1*a*, 2 and 4; and in test 3, less than one per cent. broke in the weld.

It must be noted that lap-welded tubes are rarely of uniform thickness, and variations of one to three numbers of the B. W. G. are not uncommon. This is important, especially for those whose previous experience has been with the smooth and uniform brass tubes hitherto almost entirely employed in our naval boilers, as otherwise they might be led, at first experience, to think that tubes were defective which in reality are first class in every respect. It should also be noted that the thinnest part of the tube is generally the gauge required in the boiler specifications, or, in other words, the tubes will run heavier than would appear from a calculation based on the assumption that they are of a uniform thickness equal to the specified gauge. Where a vessel is high powered, and there is a penalty for overweight, this point is worth considering.

Assistant Engineer A. M. Hunt, U. S. Navy.—In the specifications for steel-boiler tubes, drawn up by the Bureau of Steam Engineering and incorporated into Mr. Hall's article, there is

one point left untouched, which, in my opinion, deserves attention. All other boiler material is required to show not more than .035 per cent. of phosphorus, and not more than .040 per cent. of sulphur. The same requirements should be placed on the material from which tubes are made. This change was made by the Steel Inspection Board when it inserted the Bureau's specifications for tubes in the specificationss for steel for armored cruiser No. 2 and cruiser (protected) No. 6.

This change will, it is hoped, exert a beneficial influence on the pitting to which boiler tubes are so liable when made of steel. There will still be a considerable difference in the amount of carbon in the tubes and in the plates, but this cannot be avoided. Steel plates, one inch thick and over, which have a tensile strength of 58,000 to 67,000 pounds, as required by the specifications for shell plates, will necessarily have considerably higher carbon than is allowable in steel to be welded.

The English specifications for boiler tubes require tensile tests to be made on specimens cut from the finished tubes, assigning a minimum strength and elongation that will be allowed. Such a test would seem to me entirely superfluous if added to our specifications. The chemical tests compel the use of the very best material, and the ductility of the metal is fully demonstrated by tests more severe than a tensile one. It seems to me a mistake to assign a minimum limit to the tensile strength. It might compel the use of a material unsuited for welding. It is better to allow the maker to use as soft a metal as he can obtain, which is practically what he now does. The thickness of metal, necessary to resist corrosion, gives, even with the softest steel, ample strength to resist crushing and for staying tube sheets. As a matter of fact, the steel in the finished tubes has a higher tensile strength than would be expected from the low carbon. Out of twelve specimens, cut from tubes made for the Government from steel containing .09 per cent. of carbon or less, not one fell below 60,000 pounds T. S., and all gave good elongation.

XVI.

ON THE DESIGNING OF CONTINUOUS CURRENT DYNAMO MACHINES.*

By Professor Andrew Jamieson, M.Inst.C.E., F.R.S.E.

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No machine has received such constant and unremitting attention of late as the dynamo, both from men of science and from practical engineers. From being a comparatively badly designed and inefficient machine in 1880, giving then only some 50 to 60 per cent. commercial efficiency, it now reproduces over 70 per cent. of the dynamical power applied to it in the form of useful electrical power, and the electrical efficiency or ratio of electrical output to total electrical power generated often exceeds 90 per cent. Until within a very few years ago, dynamos were designed and proportioned by a rule of thumb system, or trial and failure until moderate success was attained. If a maker had luckily hit upon successful results from one machine, he copied by proportion the dimensions and materials arrived at for his other larger or smaller machines, without exactly knowing the precise causes which gave success in the one case, or failure in the other. Now all that is changed, thanks to the mathematical and experimental investigations of Dr. John Hopkinson, Professor S. P. Thompson, Mr. Gisbert Kapp, and many of the more eminent makers of dynamos, as well as to the very exact experiments by Professor Ewing on the magnetic permeabilities of different classes of iron.

If a dynamo is now required to give any desired power, the whole of its proportions and sizes can be calculated and drawn down with such accuracy that, when constructed, it will fulfil

* In large dynamos where the armature currents are great, special corrections have to be applied, which for the sake of simplicity have been neglected in the following exposition.

the stipulated conditions to within 2 or 3 per cent., with a minimum of experiment and alteration in the workshop.

Let me here add, the power is not measured by any system of "nominal power," but by actual, useful output. Engineers have of late been discussing in this Institution whether or not they should adopt one or other of the various rules for estimating the "nominal horse power" of steam engines. Nothing gives greater annoyance to an electrical engineer than having to reply to the question, "What is the nominal horse power required to drive such and such a dynamo?" He at once discards this term, and deals directly with the brake, or indirectly with the indicated horse power of the engine.

In the short time at the disposal of the author, he proposes to confine his remarks and calculations to "Continuous Current Constant Pressure Compound Dynamos," as now used for supplying current to ordinary incandescent lamps arranged upon the parallel system, or those dynamos which work in accordance with the methods illustrated by figs. 1 and 5 of the accompanying plate, where

C_a, C_m, C_s, C_e, represent the currents in the armature, main magnet coils, shunt coils, and external circuits respectively, the direction of these currents being indicated by arrows.

R_a, R_m, R_s, R_e, represent the resistances of these same parts.

E_a and E_e represent the electro-motive force or pressure generated in the armature, and the external E.M.F., or difference of pressure between the positive and negative terminals (T + and T —) respectively.

AM and VM represent current and pressure meters, or ampere and volt meters.

It is not our intention to enter here into the many mechanical and electrical arrangements of dynamos, or to trace the manner in which dynamos have been gradually simplified and cheapened. Yet as many of the members of this Institution are not familiar with the internal construction of such machines, we think that the simple illustrations (see Plate) may assist them, and lead

them to take a greater interest in the principles and formulæ which follow.

The first skeleton figure gives an idea of the general arrangement of the series Gramme dynamo, one of the few, and certainly the best machine in the market in 1880 (see fig. 2).

Fig. 3 illustrates the Faraday dynamo, as made in 1887–'88. This form of dynamo gives excellent results, but the makers, finding that it is more expensive to construct than the Kennedy dynamo, shown by fig. 4, they are now getting out a set of patterns to meet the present popular craving for cheap machines, with cast-iron core and pole pieces having but one joint, instead of a wrought-iron core and two joints.

The power given out and absorbed by dynamos is measured in watts, which is simply the product of the current in amperes, and the electrical pressure in volts. The current generated by a dynamo depends directly upon the total electrical pressure developed, and inversely upon the total resistance of the circuit (internal and external), as well as upon any back electrical pressure such as is experienced when charging accumulators or when driving motors. Neglecting this latter effect, which does not take place when merely lighting incandescent lamps,

Let E_a represent the electrical pressure generated in the armature.

E_e " external electrical pressure, or that available for useful work.

R_i " internal resistance of the whole dynamo.

R_e " external resistance of leads and lamp circuits.

C_a " current passing through armature.

C_e " current external, or that sent through lamp circuit.

Then, $$C_a = \frac{E_a}{R_i + R_e} = \text{total current generated.}$$

And, $$C_e = \frac{E_e}{R_e} \text{ external current.}$$

$$\text{The electrical efficiency} = \frac{C_e \times E_e}{C_a \times E_a} = \frac{\text{output in watts}}{\text{total watts generated}}.$$

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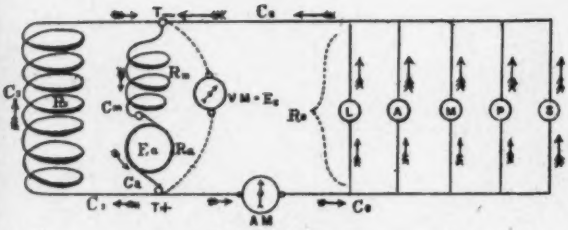
Fig. 1.



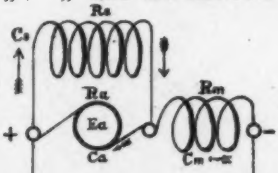
THEORETICAL DIAGRAM OF FARADAY DYNAMO CONNECTIONS

Ca, Cm, Cs, Ce represent the currents in armature, main magnet coils, shunt coils, and external circuits respectively, the direction of these currents being indicated by arrows.

Ra, Rm, Rs, Re ,, ,, *resistances of these parts.*

Ea, Ee ,, ,, EMF *in armature and on external circuit.*

Fig. 5.



THEORETICAL DIAGRAM OF RANKIN KENNEDY DYNAMO CONNECTIONS.

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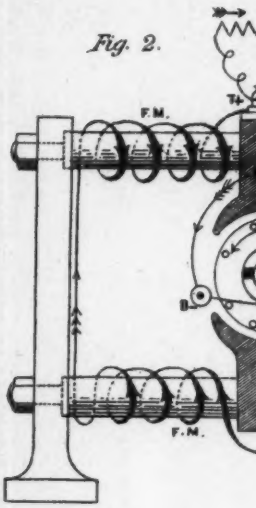
Proceedings Institution of Engineers & Shipbuilders in Scotland (Incorporated) Vol. XXXII

Fig. 2.



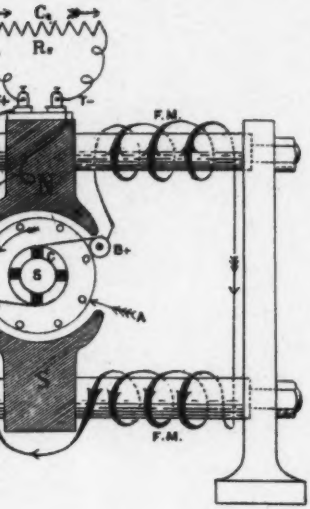
GRAMME DYNAMO, s

The Armature A *is shown rev*
position in the Machine,

INDEX

A for Arm
S ,, Spi
CC ,, Com
B+ ,, Posi
B- ,, Neg
T- ,,
T+ ,, Posi
FM ,, Fiel
NS ,, Nort
Re ,, Resi
Ce ,, Curr

Fig. 6.



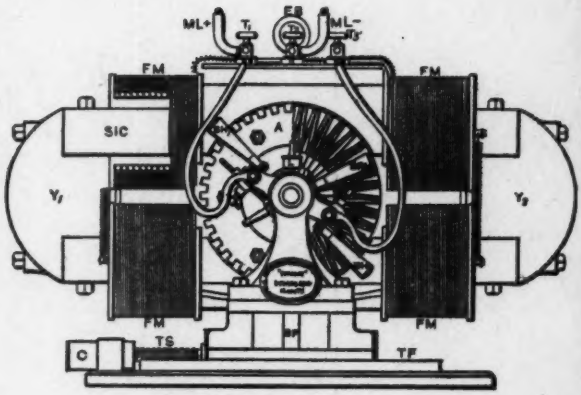
, showing Connections, &c.

revolving at right angles to its natural
ne, for simplicity of explanation.

EX TO PARTS.

Armature.
Spindle.
Commutator.
Positive Brush.
Negative ,,
,, Terminal.
Positive ,,
Field Magnets.
North & South Poles
Resistance External
Current ,,

Fig. 3.



FARADAY DYNAMO.

INDEX TO PARTS.

BP	-	represents the Bed Plate.
TF	-	,, Tightening Frame.
TS	-	,, ,, Screw.
FM	-	,, Field-Magnets.
Y_1, Y_2	-	,, ,, ,, Yokes (cast-iron).
SIC	-	,, Soft Iron Cores.
A	-	,, Armature.
BH_1, BH_2		,, Brush Handles.
ML+ ML−		,, Main Leads (positive and negative).
T_1 T_2 T_3		,, Terminals

Fig. 7.

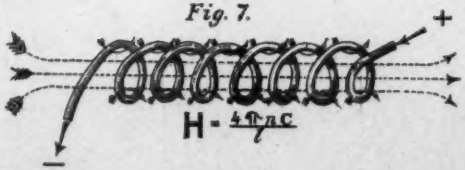
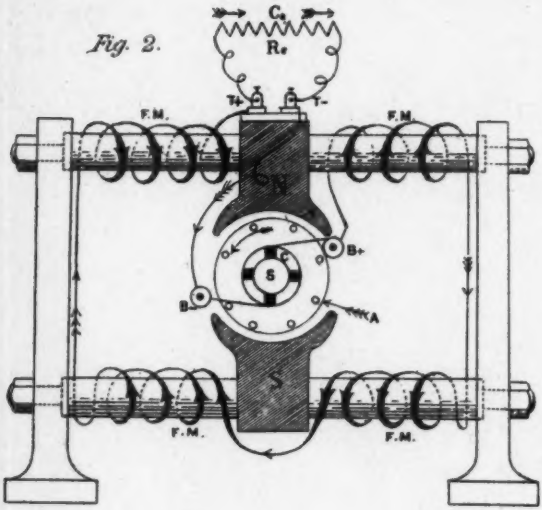


Fig. 2.



GRAMME DYNAMO, showing Connections, &c.

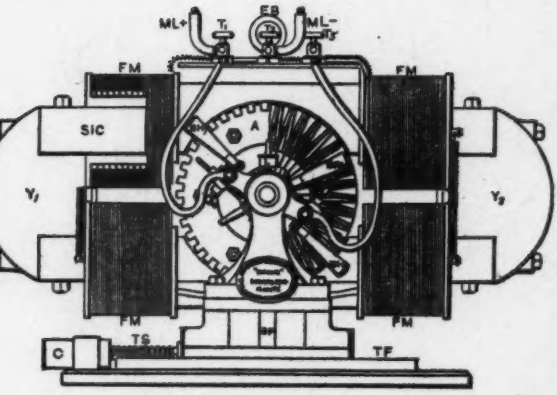
The Armature A *is shown revolving at right angles to its natural position in the Machine, for simplicity of explanation.*

INDEX TO PARTS.

A	for	Armature.
S	„	Spindle.
CC	„	Commutator.
B+	„	Positive Brush.
B-	„	Negative „
T-	„	„ Terminal.
T+	„	Positive „
FM	„	Field Magnets.
NS	.	North & South Poles
Re	.	Resistance External
Ce	.	Current „

Fig. 6.

Fig. 3.

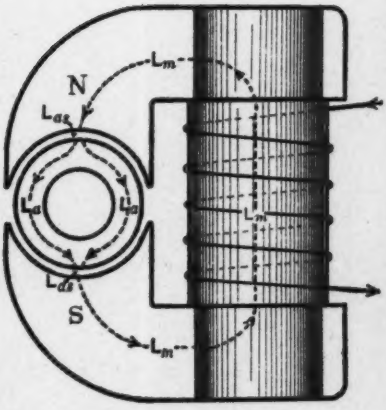


FARADAY DYNAMO.

INDEX TO PARTS.

BP	-	represents the Bed Plate.
TF	-	„ Tightening Frame.
TS	-	„ „ Screw.
FM	-	„ Field-Magnets.
Y_1, Y_2	-	„ „ „ Yokes (cast-iron).
SIC	-	„ Soft Iron Cores.
A	-	„ Armature.
BH_1, BH_2		„ Brush Handles.
ML+ ML−		„ Main Leads (positive and negative).
T_1 T_2 T_3		„ Terminals

Fig. 4.



MAGNETIC CIRCUIT OF SINGLE BOBBIN HORSE-SHOE KENNEDY DYNAMO.

L_m *represents length of field magnet circuit.*

L_{as} „ *length of air space circuit.*

L_a „ *length of armature circuit.*

Fig. 7.

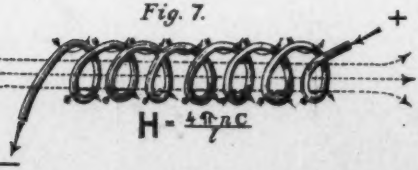
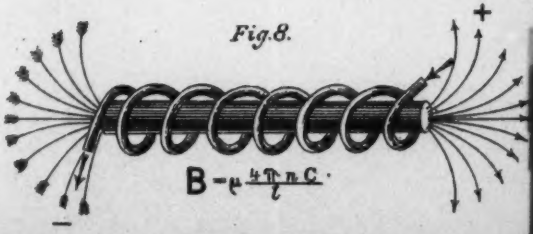


Fig. 8.



Now, since C_a, E_e, and C_e depend directly upon E_a, we propose to show how this internal electrical pressure may be obtained for any dynamo of the kind under consideration.

This total or internal pressure, E_a, depends directly upon—

1. The total number of magnetic lines which flow between the field magnet poles through the armature.
2. The number of convolutions of the armature conductors which are in series at one time.
3. The speed of rotation of the armature.

The *number of revolutions* per minute is a matter that can be easily decided upon before beginning to design the dynamo. Fast speed dynamos, if well balanced, are common in land installations; slow speed ones on board ship, and when vibration and noise must be avoided.

The *number of convolutions* of the armature conductor depends upon the diameter of the armature, the number of layers on armature or depth of winding, and the size of wire to be used, all of which will be determined as the drawing or design proceeds in accordance with sound practice.

The *magnetic circuit* is the chief and most intricate part of the whole design which requires our attention, and we shall, therefore, consider it first in full detail.

I. FIELD MAGNET CIRCUIT.

The field magnet circuit should have as little magnetic resistance as possible. The magnet cores, &c., *should* therefore be made of the best soft Swedish iron, short, of large sectional area, and with no joints. Owing, however, to expense, the difficulty of winding the wire on the magnets, or the fixing of bobbins containing the wire, and the necessity of having sufficient residual magnetism with which to start the dynamo current, and to prevent fluctuations of current, the yoke (if there is one) and the pole pieces are frequently made of cast-iron.

In a Kennedy dynamo (see fig. 4) the mean length of the magnetic circuit is indicated by the letter L_m, and the poles are marked N and S.

II. AIR SPACES.

The air spaces should have the least possible magnetic resistance. They should, therefore, have as little depth as is consistent with preventing actual contact between the armature wires and the pole pieces, and of allowing free circulation of air, and of preventing heating of the pole pieces due to Foucault or eddy currents in the forward ends of the pole pieces.

In fig. 4 the two air spaces are marked, L_{as}, L_{as}.

III. ARMATURE CORE.

The armature core should have as little magnetic resistance as possible, and be as free as possible from inducing Foucault currents. It should, therefore, be built up of the softest thin Swedish iron plates, well insulated from each other, and laminated or placed in a direction parallel to the direction of the magnetic lines passing between the poles of the field magnet. Some makers prefer to build up their armature cores of soft iron wire insulated by the natural thin coating of oxide of iron always found on such wire.

In fig. 4 the mean directions of the split magnetic circuit through the armature core are marked, L_a, and, L_a.

IV. MAGNETIC RESISTANCE OF FIELD MAGNET CIRCUIT.

The magnet circuit, L_m, has a magnetic resistance.

$$R_m = 2\frac{L_m}{S_m}.$$

Where—

2 = Kapp's co-efficient for the *initial specific* magnetic resistance of soft wrought-iron (the number 3 to be used if core, &c., be of cast-iron).

L_m = the mean length of field magnet circuit in inches.

S_m = sectional area in square inches (at right angles to L_m) of magnet core.

V. MAGNETIC RESISTANCE OF AIR SPACES.

The air spaces circuits, L_{as}, L_{as}, have a resistance,

$$R_{as} = 1440 \frac{2 L_{as}}{S_{as}}.$$

Where—

1440 = Kapp's co-efficient for the *specific* magnetic resistance of air and copper when wrought-iron magnets are used (1800 to be used with cast-iron magnet cores and pole pieces, and 1500 with wrought-iron cores and cast-iron poles).

L_{as} = depth of each air space in inches (and since there are two spaces, 2 L_{as} is inserted in the formula.)

S_{as} = sectional area of each air space or surface area of each pole piece in square inches.

VI. MAGNETIC RESISTANCE OF ARMATURE CORE.

The armature core circuits, L_a, L_a, have a magnetic resistance.

$$R_a = 2 \frac{L_a}{S_a}.$$

Where—

L_a = mean average length of armature core magnetic circuit in inches.

2 = Kapp's co-efficient for wrought-iron.

S_a = total cross sectional area of armature core in square inches = twice radial depth × length of armature core.

VII. TOTAL MAGNETIC RESISTANCE.

The total magnetic resistance of the complete magnetic circuit is therefore—

$$R_m + R_{as} + R_a = 2 \frac{L_m}{S_m} + 1440 \frac{2 L_{as}}{S_{as}} + \frac{2 L_a}{S_a}.$$

VIII. KAPP LINES.*

The total number of Kapp lines, K_L, of magnetic force which pass through the magnet and air space, can therefore be calculated by *dividing* the ampere-turns ($A \times T_{mw}$) on the magnet windings by the total magnetic resistance in the magnetic circuit.

Or
$$K_L = \frac{A \times T_{mw}}{R_m + R_{ax} + R_a}.$$

i. e.,
$$K_L = \frac{A + T_{mw}}{2\,\frac{L_m}{S_m} + 1440\,\frac{2\,L_{ax}}{S_{ax}} \times 2\,\frac{L_a}{S_a}}.$$

IX. E.M.F. IN ARMATURE.

The E.M.F. developed internally in the armature winding is—
$$E_a = K_{La} \times T_{aw} \times N_{ar} \times 10^{-6}.$$

Where—

E_a = E.M.F. developed in armature in volts.

K_{La} = Kapp lines which only pass through armature or the useful field.

---

*A Kapp line is simply = 6000 C.G.S. magnetic lines. Kapp proposed this unit (see *Journal Society Telegraph Engineers and Electricians*, vol. xv, p. 518, 1886,) as a suitable factory unit for dynamo use, on account of the C.G.S. unit being so small; also from the fact that the revolutions of dynamo armatures are usually reckoned per minute instead of per second, and that British workmen still measure sectional areas in square inches instead of square centimeters, as in the C.G.S. system.

The number 6000 is made up of two terms—viz., 60, which enables revolutions to be counted per minute instead of per second; and 100, to bring down the measurement of the magnetic lines to reasonable numbers; or $\frac{\text{C.G.S. lines}}{60 \times 100} = \frac{\text{C.G.S. lines}}{6000}$.

Then, 1 Kapp line = 6000 C.G.S. lines.

Since there are 6.4514 centimeters in a square inch,

1 Kapp line per sq. inch $= \frac{6000}{6.4514} = 930$ C.G.S. lines per square centimeter. Consequently, instead of saying 18,600 C.G.S. lines, we simply say 20 Kapp lines, or $\left(\frac{18{,}600}{930}\right)$, which gives small numbers more easily expressed and understood by those engaged in dynamo construction; in the same way that for long distances we prefer to measure lengths in miles instead of in yards or in inches. (See page 326 for continuation of explanation.)

T_{ae} = Turns of armature wires in series (counted all round core).

N_{ar} = number of armature revolutions *per minute*.

EXAMPLE I.*—Let us now apply the foregoing formulæ to the design of a single field compound dynamo of similar form to that shown in figs. 4 and 5.

Let the dynamo be furnished with a long shunt, and be of 6400 watts (external power P_w), when driven at 1000 revolutions per minute.

First, let us determine the resistance, current, fall of potential, in armature and magnets, and percentage loss of power in the armature, shunt, and main field magnet coils, using the same symbols as before.

Given $\left.\begin{matrix} E_e = 100 \text{ volts.} \\ C_e = 64 \text{ amperes.} \end{matrix}\right\} \therefore E_e \times C_e = 6400\ P_w.$

Resistances.—Then $R_e = \dfrac{E_e}{C_e} = \dfrac{100\ v}{64\ a} = 1.55$ ohms.

Suppose we agree to a total loss of about 10 per cent. of the total power in the dynamo, or say—

The electrical efficiency of the dynamo = 90 per cent.

$\therefore$ 90 per cent. : 100 per cent. :: 6400 $P_w : x$.

$\therefore$ Total power $= x = 7100\ P_w$.

Let $R_a = \dfrac{R_e}{20} = \dfrac{1.55}{20} = .077\ \omega.$

Let $R_s = R_a \times 1000 = .077 \times 1000 = 77\ \omega.$

Let $R_m = 75\ \% R_a = \frac{3}{4} \times .077 = .058\ \omega.$

Now, the external current = current in armature and field magnet main coils less the current in shunt.

Currents.—Or $C_e = C_a - C_s.$

But, $C_s = \dfrac{E_e}{R_s} = \dfrac{100\ v}{77\ \omega} = 1.3\ a$

$\therefore 64\ a = C_a - 1.3\ a.$

$C_a = 64 + 1.3.$

$= 65.3$ amperes.

* The calculations for Example I. were made by Boucher's Calculator, and thus are therefore only approximately true.—A. J.

Fall of potential in armature $= C_a \cdot R_a = 65.3 \times 0.77$
$= 5.02$ volts.

Fall of potential in magnet main coils $= C_m \cdot R_m = 65.3 \times .058$
$= 3.78$ volts.

Total fall of potential in armature and magnet series coils
$= C_a$ or $C_m (R_a + R_m)$.
$= 65.3 (.077 + .058) = 8.8\ v.$
$\therefore E_a = 8.8 + E_v = 108.8$ *volts.*

Percentage losses of power.—

In armature $= C_a^2 \cdot R_a = 65.3^2 \times .077$
$= 330\ P_w.$
$\therefore 7100\ P_w : 330\ P_w :: 100 : x_a$
$x_a = 4.63$ per cent.

In shunt coils $= C_s^2 \cdot R_s = 1.3_s^2 \times 77\ \omega$
$= 132\ P_w.$
$\therefore 7100\ P_w : 132\ P_w :: 100 : x_s$
$x_s = 1.86$ per cent.

In main coils $= C_m^2 \cdot R_m = 65.3^2 \times .058$
$= 248\ P_w.$
$\therefore 7100 : 248 :: 100 : x_m.$
$x_m = 3.5$ per cent.

Totalling the different percentage losses we have:—

In armature,	- -	$x_a = 4.63$ per cent.
In shunt coils,	- -	$x_s = 1.86$ "
In main coils,	- -	$x_m = 3.5$ "
		9.99 %, or say 10 %.

Kapp Lines in Armature Core.—

Now, before substituting the above data in Kapp's formula, we must agree upon the turns of wire in the armature at the agreed upon 1000 revolutions per minute. Let there be 36 commutator bars or 36 sections of wire in armature, which has been found sufficient to give a current of practically uniform E.M.F. The number of turns of wire in each section of the armature depends upon whether we use double or single winding, and upon the current and the resistance of the armature wire. As

we shall see further on, from our trial drawing and calculations, we can get in 6 turns of wire to each section.

$$\therefore T_{aw} = 36 \times 6 = 216 \text{ turns of armature wire.}$$

Let $N_{ar} = 1000$ revolutions per minute.

Now $$E_v = K_{Le} \times T_{aw} \times N_{ar} \times 10^{-6}.$$

$$\therefore K_{Le} = \frac{E_v}{T_{aw} \times N_{ar} \times 10^{-6}}$$

$$= \frac{108.8\ v}{216 \times 1000 \times 10^{-6}} = 503.7.$$

say = 500 Kapp lines in *armature core*.

In order to obtain a certain number of Kapp lines or magnetic induction in an armature core, it is necessary to give a greater magnetic intensity to the field magnet core or cores, owing to the unavoidable leakage of magnetic lines which do not pass through the armature core.

Dimensions of Magnet Core.—

In ordinary practice it is usual to assume that only about 80 per cent. of the field magnet lines pass through the armature core.

Therefore, the Kapp lines in *Field Magnet* are,

$$80 \text{ per cent.} : 100 \text{ per cent.} :: 500 : x = 625.$$

We must now ascertain the dimensions of the field magnet core to carry these 625 Kapp lines. Suppose we consider the case of field magnets to be cast-iron throughout (poles and core).

A safe working density is 9 Kapp lines per square inch,

$$\therefore S_m = \frac{625}{9} = 69.4 \text{ square inches.}$$

$$= 9.4 \text{ inches diameter (if round).}$$

In practice it is found that a dynamo magnet bobbin should be in length about $1\frac{1}{4}$ times its diameter to give the necessary space for winding or ampere-turns.

$$\therefore 9.4 \times 1.25 = 11.75 \text{ inches long.}$$

Dimensions of Armature Core.—

It has been found advisable, in designing the dimensions of the armature core of a dynamo having a field magnet of the form we are now considering, to make the diameter of its core slightly

larger, and its length slightly less, than the diameter of the field magnet core.

Therefore, calling the diameter of the Gramme armature core 10 inches, its length 9 inches, and allowing a density of 20 Kapp lines per square inch of core section, since it is made of the best permeable wrought-iron,

We have $\frac{500}{20} = 25$ square inches total cross section of armature.

$$\therefore S_a = 25 \text{ square inches.}$$

Allowing 20 per cent. of total space for insulation between *laminated plates* in armature core, we have

80 per cent. : 100 per cent. :: $25 : x = 31.5$ square inches.

$$\therefore \frac{31.5}{2} = 15.75 \text{ square inches in each half.}$$

$$\therefore \frac{15.75}{9''} = 1.75 \text{ inches} = \text{radial depth of core.}$$

Magnetic Resistance of Field Magnet.—

Now, drawing armature and field magnets to scale (as per fig. 4), and allowing $2\frac{1}{2}$ inches for depth of winding on field magnet bobbin and clearance for armature, we find the mean length of field magnet core, $L_m = 44$ inches, and remembering that in a formula for the magnetic resistance of this part of the circuit we have to use a co-efficient of 3 instead of 2, seeing that the core is of cast-iron,

We have $$R_m = 3 \frac{L_m}{S_m}.$$

$$= 3 \frac{44}{69.4} = 1.9.$$

Magnetic Resistance of Air Space.—

With *two* layers of insulated conductors having a cross section (▭) of .08 inch deep, by .2 inch broad (including covering), we find the total depth of air space on one side between pole pieces and armature core (allowing .14 inch mechanical clearance), equal .3 of an inch. Consequently, $2\ L_{as} = 2 \times .3 = .6$ inches. The bore of pole pieces will be = diameter of armature core + $2\ L_{as}$

= 10 inches + .6 inches = 10.6 inches.

It is good practice to allow the corners of pole pieces to form an angle of 30° with center of armature. The angle embraced by each pole piece will therefore be 180° — 30° = 150°, which gives us a circumferential length of say 14 inches for each pole. And since our armature core is 9 inches long, and it is unusual to make the pole pieces broader than this, we have

$$S_{as} = 14 \text{ inches} \times 9 \text{ inches} = 126 \text{ square inches.}$$

Hence—

$$R_{as} = 1800 \frac{2 L_{as}}{S_{as}},$$

$$= 1800 \frac{2 \times .3}{126} = 8.5.$$

Magnetic Resistance of Armature.—

Referring to scale or full-size section, we find by measurement that, $L_a = 15$ inches. We have already determined S_a to be 25 square inches.

Consequently—

$$R_a = 2 \frac{L_a}{S_a},$$

$$= \frac{2 \times 15}{25} = 1.2.$$

$$\therefore R_m + R_{as} + R_a = 1.9 + 8.5 + 1.2.$$

say = 11 Kapp units of magnetic resistance.

Ampere Turns.—From formula for *cast*-iron, under heading VIII

$$K_L = \frac{A \times T_{mw}}{R_m + R_{as} + R_a} = 625 \text{ (gross Kapp lines)}$$

$$\therefore A \times T_{mw} = K_L (R_m + R_{as} + R_a),$$

$$= 625 \times 11,$$

$$= 6875 \text{ total ampere-turns in shunt and main field magnet coils.}$$

Proportioning of Ampere-Turns between Shunt and Main Coils.—

The total ampere-turns have now to be divided in proper proportion between the series coils and the shunt coils of the field magnet windings.

In order to do this, so as to give the desired form of characteristic curve, we must have recourse to actual experiment. The following method is carried out by Messrs. McWhirter, Ferguson & Co., Faraday Works, Govan.

1. A workshop bobbin or experimental coil, with a known number of turns of wire, is kept for every size of dynamo machine made. This bobbin is put on to the magnet core in circuit with an ampere-meter and is separately excited by another dynamo or storage cells. The dynamo to be wound is run at the normal speed with the external circuit, R_e, equal to that of shunt, viz., 77 ω, and with the voltmeter placed between the terminals. The exciting current through the experimental bobbin is now gradually increased until the required number of volts is observed on the voltmeter. *E.g.*, suppose the experimental bobbin to have 500 turns, and the ampere-meter reads 8.25 amperes, when the normal voltage of 100 volts is reached, then the ampere-turns of shunt wire

$$A \times T_{sw} = 500 \times 8.25 = 4125 \text{ ampere-turns.}$$

2. Connect up the armature of the dynamo being constructed to a resistance equal to the joint resistance of the shunt and the normal lamp circuit through an ampere-meter, and run the dynamo at the normal speed. Now increase the exciting current in the experimental bobbin until the normal voltage of 100 volts and normal armature current of 65.3 amperes are both obtained. At the same time observe the exciting current.

But, the total required ampere-turns = 6875, and from experiment (1) we found that 4125 ampere-turns produced the requisite E.M.F.

$\therefore 6875 - 4125 = 2750$ ampere-turns left for series coils.

3. The normal current at full load required to pass through the series coil was found to be 65.3 amperes.

Consequently the number of turns required in the series coils

$$= \frac{2750}{65.3} = 42 \text{ (about).}$$

4. From experiment (1), we found that 4125 ampere-turns gave us the required excitation to produce the normal E.M.F., and from our former calculation, we found that the current in shunt,

$$C_s = 1.3 \text{ amperes.}$$

$$\therefore \frac{4125}{1.3} = 3173 \text{ turns.}$$

But the series coils are also carrying the 1.3 amperes required for the shunt when dynamo free.

$\therefore 3173 - (42 \text{ turns} \times 1.3\,a) = 3173 - 54 = 3119$ turns on shunt coils.

5. All we now have to do is, to so proportion the size of our series and shunt coil conductors that they shall carry the necessary currents required in each case at the rate of not exceeding 1000 amperes per square inch. For armatures this may be and is in practice increased to even 3000 amperes per square inch without undue overheating, owing to the cooling effect of the air carrying off the heat generated in the revolving armature. Mere radiation only acts on the field magnet coils; therefore, the sectional area is of necessity much greater.

EXAMPLE II.—Now take the case of a Rankin-Kennedy short shunt compound dynamo, as shown by figs. 4 and 5. The particular dynamo in question was tested by the writer after completion with the following results:—

Resistance of armature, warm,	$R_a = .04$ ohm.
Resistance of magnet shunt, warm,	$R_s = 20$ ohms.
Resistance of magnet main coil, warm,	$R_m = .03$ ohm.
Current in working circuit,	$C_e = 108.5$ amperes.
Current in shunt magnet coils,	$C_s = 5$ amperes.
Difference of potential at dynamo terminals,	$E_e = 100$ volts.
Difference of potential at brushes,	$E_b = 103$ volts.
E.M.F. generated in armature,	$E_a = 108$ volts.
Turns of armature wire,	$T_{aw} = 180$.
Number of armature revolutions per minute,	$N_{ar} = 620$.

ELECTRICAL EFFICIENCIES.

External electrical power, or "output," available for electric lighting, &c., $= C_e \times E_e = 108.5_a \times 100$ v $= 10{,}850$ watts $= 88.7\%$

Power absorbed by armature,

$= C_a^2 \times R_a = (113.5)_a^2 \times .04\,\omega = 515$ watts $= 4.2\%$

Power absorbed by shunt magnet coils,

$= C_s \times E_s = 5_a \times 103$ v $= 515$ watts $= 4.2\%$

Power absorbed by main magnet coils,

$$= C_m^2 \times R_m = (108.5)^2_8 \times .03\omega = 353 \text{ watts} = 2.9\%$$

Total electrical power,

$$(C_e \times E_e) + (C_a^2 \times R_a) + (C_s \times E_s) + (C_m^2 \times R_m) = 12{,}233 \text{ watts.}$$

The only difference between this dynamo and our last example being that the magnet core was made of the best hammered scrap-iron instead of cast-iron, and consequently the permeability is greater. The constants to be used in the formula are therefore less, and we have to substitute 2 for 3, and 1500 for 1800 as found by experiment.

From the measured dimensions of the dynamo we found:—

$L_m = 38$ inches.	$S_m = 98$ square inches.
$L_{as} = 0.45$ "	$S_{as} = 195$ "
$L_a = 15$ "	$S_a = 46$ "

$$R_m = 2\frac{L_m}{S_m} = 2\frac{38}{98} = .76.$$

$$R_{as} = 1500\frac{2\,L_{as}}{S_{as}} = 1500\frac{2 \times .45}{195} = 6.9.$$

$$R_a = 2\frac{L_a}{S_a} = \frac{2 \times 15}{46} = .65.$$

Total magnetic resistance = 8.31.

Now by substituting in our former formula the actual values obtained for the E.M.F. in the armature, turns of armature wire, and revolutions per minute, we get the magnetic flux in Kapp lines flowing through the armature. Or,

$$K_{L_a} = \frac{E_a}{T_{ae} \times N_{ar} \times 10^{-6}}$$

$$= \frac{108}{180 \times 620 \times 10^{-6}}$$

$$= 968 \textit{ lines of magnetisation through armature, or at the rate of } 21 \textit{ Kapp lines per square inch.}$$

But in dynamos of this class it has been found that the number of magnetic lines passing through the armature are only 80

per cent. of the total magnetisation in the field magnets. Consequently,

$$80\ \% : 100\ \% :: 968 : x$$

$\therefore x = 1210$ *Kapp lines through the magnet cores, or at the rate of* 12.3 *Kapp lines per square inch.*

From our former formula,

$$K_L = \frac{A \times T_{mw}}{R_m + R_{as} + R_a}$$

$$1210 = \frac{A \times T_{mw}}{8.31}$$

$= 10{,}055$ ampere-turns on the field magnets.

The actual ampere-turns were 9800, or the calculation is only about 3 per cent. more than the *actual* amount required to satisfy the requirements of the installation.

EXAMPLE III.—Take the case of a Faraday compound dynamo, also tested by the writer, supplied by Messrs. McWhirter, Ferguson & Co., Govan, to the Blair Lodge School. Where (see figs. 1 and 3),

$R_a = .0315\ \omega$	$E_e = 100$ v
$R_s = 31.0\ \omega$	$E_a = 107.6$ v
$R_m = 0.012\ \omega$	$E_b = 102$ v
$C_e = 130.0\ a$	$T_{av} = 240$ (48 commutator bars)
$C_a = 3.55\ a$	$N_{av} = 800$

This dynamo being of the double horse-shoe magnet type, and the cores being of the best Swedish iron, with large cast-iron yokes we use a constant 2 for wrought-iron cores, and 1440 for specific resistance of copper and air space due to wrought cores and pole pieces.

From the actual dimensions of the dynamo we found :—

$L_m = 32.0$ inches.	$S_m = 33$ square inches (only 1 magnet core).
$L_{as} = 0.5$ "	$S_{as} = 54$ " ($\frac{1}{2}$ air space surface).
$L_a = 12.0$ "	$S_a = 15.6$ " ($\frac{1}{2}$ section of armature).

$$\therefore R_m = 2\frac{L_m}{S_m} = 2\frac{32}{33} = 1.94$$

$$R_{as} = 1440 \frac{2 L_{as}}{S_{as}} = 1440 \frac{2 \times .5}{54} = 26.6$$

$$R_a = 2 \frac{L_a}{S_a} = 2 \frac{12}{15.6} = 1.54$$

Magnetic resistance of half-dynamo field = 30.08

Now,

$$K_{La} = \frac{E_a}{T_{aw} \times N_{ar} \times 10^{-6}}$$

$$= \frac{107.6}{240 \times 800 \times 10^{-6}}$$

= 530 *lines of magnetisation through armature, or at the rate of* $\frac{530}{15.6 \times 2} = 17$ *Kapp lines per square inch of armature section.*

Allowing, as before, that only 80 per cent. of the total magnetisation passes through the armature, we have

$$80\,\% : 100\,\% :: 530 : x.$$

$x = 662$ Kapp lines through both the magnet cores, or 331 lines through each half, which is at the rate of 10 Kapp lines per square inch of magnet core section.

$$\therefore K_L = \frac{A \times T_{mw}}{R_m + R_{as} + R_a}$$

$$\text{Or } A \times T_{mw} = K_L (R_m + R_{as} + R_a)$$

$$\therefore A \times T_{mw} = 331 \times 2 \times 30.08 = 19{,}913$$

The actual ampere-turns were 20,850, and the dynamo having a rising characteristic of 3 per cent., the calculation is only about 1.7 per cent. less than actually found by experiment.

The discussion on this paper took place on 26th February, 1889.

Professor JAMIESON explained the leading features of his paper by referring to the wall diagrams which he had prepared of the magnetic and electrical circuits of the Kennedy and Faraday dynamos. He concluded by working out the following demonstration of how magnetic fields are measured, and of how Kapp's

constants are derived. He also stated what changes had to be made in these constants to suit different degrees of magnetisation and qualities of iron.

Unit Poles.—The force, F, exerted between two poles of strengths m and m', placed at a distance, L, centimeters from each other, is

$$F = \frac{m \times m'}{L^2} \text{ dynes.}$$

If m and m' are each unit poles, and their distance apart is 1 centimeter, then

$$F = \frac{m \times m'}{L^2} = \frac{1 \times 1}{1 \times 1} = 1 \text{ dyne of force.}$$

A dyne is unit force in the C.G.S. system, or that force which if acting for 1 second on a mass of 1 gramme, imparts to it a velocity of 1 centimeter per second.

Method of Reckoning the Number of Magnetic Lines of Force in a Magnetic Field.—

Suppose a free unit pole, p, situated at the center of a non-magnetic sphere of unit radius, r (1 centimeter), and that the whole surface of this sphere is plotted out into squares of unit area (1 square centimeter), (see fig. 6). Further, suppose that free unit poles, p_1, p_2 . . . p_{12}, are placed at the centers of *each* of these squares, then it follows from the previous statements that the unit pole at the center of the sphere would exert unit force (or one dyne) on each of the unit poles at the surface of the sphere.

But the surface of a sphere $= 4 \pi r^2$.

and since $r = 1$ centimeter or unit length, the surface of the sphere is $= 4 \pi$ square centimeters; therefore, there are 4π dynes of force emanating from the central unit pole, or what is usually termed 4π magnetic lines. Consequently a magnetic line represents a force of one dyne.

If we have any other magnetic pole of strength, M, then the number of lines which flow from it are equal to

$$4 \pi \text{ M lines.}$$

Let M = 100, then

$$4 \pi \text{ M} = 4 \times 3.1416 \times 100 = 1257 \text{ lines.}$$

Magnetic Fields Produced by Currents in Solenoids.—

Referring again to the Plate, we see fig. 8 representing a spiral conductor or solenoid (without any iron or other core) conveying a current which produces a magnetic field. Then the intensity or number of lines per square centimeter of its cross section (at right angles to the direction of the lines and at the center) is equal to

$$\mathsf{H} = \frac{4\pi n \mathsf{C}}{l}$$

(see index to symbols, below).

In fig. 9 the same solenoid is fitted with an iron core, and the same current is supposed to be sent through it. Now, however, we find that the number of magnetic lines per square centimeter of cross section (within the bar) have increased to

$$\mathsf{B} = \mu \mathsf{H} = \mu \frac{4\pi n \mathsf{C}}{l}.$$

1. *Method of deriving Kapp's constant, 2, from the fundamental formulæ for* H *and* B.—

Let H stand for intensity of field, or C.G.S. lines per square centimeter.

" B " magnetisation induced by H in the iron in C.G.S. lines per square centimeter.

" μ " permeability of the iron core.

" C " current in solenoid in C.G.S. units.

" n " number of turns of spirals in solenoid.

" l " length of solenoid in centimeters = 2.5, L (inches).

" s " cross area of iron in sq. cms. = 6.45, S (sq.ins.).

$$\mu = \frac{\mathsf{B}}{\mathsf{H}}, \qquad \therefore \mathsf{B} = \mu \mathsf{H}.$$

But, $$\mathsf{H} = \frac{4\pi n \mathsf{C}}{l}, \qquad \therefore \mathsf{B} = \mu \frac{4\pi n \mathsf{C}}{l}.$$

Now, as stated before, 1 Kapp line = 6000 C.G.S. lines (see p. 314).

$\therefore$ The Kapp lines induced in the core by current in solenoid $= \dfrac{\text{the number of C.G.S. lines per sq. in.} \times \text{area of core in sq. ins.}}{6000}$

$$K_L = \frac{B \times \text{area in sq. inches}}{6000} = \mu \frac{4\pi n C \times 6.45}{6000 \times 2.5} \times \frac{\text{sq. inches}}{\text{inches}}$$

$$" = \mu \frac{4\pi 10^{-1} A \times T_{mw} \times 6.45\, S}{6000 \times 2.5\, L} = \mu \frac{3.19 \times \text{exciting force} \times S}{6000 \times L}.$$

But the magnetic resistance

$$R = \frac{\text{exciting force}}{\text{magnetisation}} = \frac{A \times T_{mw}}{\mu \frac{3.19\, A \times T_{mw} \times S}{6000 \times L}}$$

$$\therefore R = \frac{6000\, L}{3.19\, \mu\, S} = 1860 \frac{L}{\mu\, S}.$$

Now, where $\mu = 930$ for magnet wrought-iron, as found by experiment, or at the rate of about 14 Kapp lines per sq. inch,

$$R_m = \frac{1860\, L_m}{930\, S_m} = 2 \frac{L_m}{S_m}.$$

N. B.—For other values of, μ, the co-efficient, 2, varies accordingly.

2. *Precisely in the same way the constant* 1440 *is found for the air space.* For,

$$R_{as} = \frac{6000\, L_{as}}{3.19\, \mu\, S_{as}} = 1860 \frac{L_{as}}{\mu\, S_{as}}.$$

Now, if the air space was simply filled with air alone, μ would be equal to 1, and 1860 would be the constant; but it is to a large extent filled by the insulated copper wire of the armature coils, and Mr. Kapp has found, from an average of a number of experiments, that this increases the permeability, μ, to 1.36.

Consequently,

$$R_{as} = \frac{1860\, L_{as}}{1.36\, S_{as}} = 1440 \frac{L_{as}}{S_{as}}.$$

N. B.—For other values of, μ the constant, 1440, varies accordingly.

3. *Tangent Function Method for constant,* 2.

Iron molecules when unmagnetised are supposed to lie so, 00
When about half saturated they are supposed to be twisted by the magnetic force through an angle so, - - - - 00
When completely saturated they are supposed to have been turned through 90° from original position, so, - - - 00

Kapp consequently assumes that the molecules of dynamo iron cores have been twisted through 90° when completely saturated with magnetism, or that the tangent of 90° (which is equal to infinity) represents the magnetic resistance in this extreme condition, and the tangent of some intermediate angle represents the magnetic resistance of the actual working magnetisation.

Let σ_m stand for the efficiency of the working magnetisation, or ratio of working magnetisation to complete magnetisation.

Then if 18 Kapp lines per square inch be the working magnetisation of best wrought-iron, and 26 Kapp lines per square inch be complete saturation of the iron, whilst the initial magnetic resistance = 1.

Then

$$\sigma_m = \frac{\text{K lines per sq. in. working magnetisation}}{\text{K lines per sq. in. complete magnetisation}} = \frac{18}{26}$$

$$= 69\ \%\ \text{saturation.}$$

$\therefore$ The magnetic resistance R_m

$$= \frac{\text{initial magnetic res} \times \tan \frac{\pi}{2}\sigma_m}{\frac{\pi}{2}\sigma_m}.$$

$$R_m = \frac{1 \times \tan 90° \frac{18}{26}}{\frac{3.1416}{2} \times \frac{18}{26}} = \frac{\tan 62°}{1.049} = \frac{2.13}{1.049} = 2 \text{ (nearly).}$$

From the foregoing it will be observed that Kapp's constants only hold good for particular permeabilities of the different parts of the magnetic circuit.* Nevertheless, the formula is an extremely convenient one, and as has been proved by the examples

* See Electric Transmission of Energy, 2d edition, page 121, by Gisbert Kapp, (publishers: Messrs. Whittaker & Co., London,) or The Electrical Trades Directory, 1890 edition, page 148, for Snell's Tables of Tangent Function Multipliers to be used with Kapp's formula in the designing of dynamos.

given, it may be trusted as far as it goes. To those who prefer to refer to tables of magnetic permeabilities, and to work with centimeters, grammes, and seconds, the following formula may be used, as more scientifically accurate and elastic in its application.*

The Ampere Turns on Magnet Coils.—

$$A \times T_{mw} = M_a \frac{\left\{ \lambda \frac{L_m}{S_m \cdot \mu_m} + \frac{2 L_{as}}{S_{as} \cdot \mu_{as}} + \frac{L_a}{S_a \cdot \mu_a} \right.}{4 \pi 10^{-1}}.$$

Where M_a stands for magnetisation of armature in C.G.S. lines.

λ " the leakage co-efficient mentioned before, or λM_a = the magnetisation of the field magnets, *i.e.*, M_a is less than magnetisation of magnets by say 20 per cent.

$\mu_m \mu_{as} \mu_a$ " the permeabilities of field magnet iron, air space, and armature core, to be obtained from tests of these or from tables.

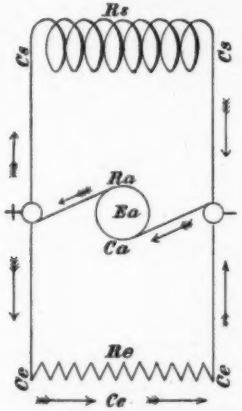
$L_m L_{as} L_a$ " the same as before, but in centimeters.

$S_m S_{as} S_a$ " the same as before, but in square centimeters.

The object of the paper has, however, been to present the calculations and examples in as simple a form as possible, so that they might be understood and applied by any engineer or draughtsman not accustomed to the centimeter-gramme-second system of units.

* See section on Dynamos by Professor S. P. Thompson, D.Sc., in the sixth edition of Munroe and Jamieson's pocket book of Electrical Rules and Tables.

EXAMPLE IV.—Now take the case of a Faraday dynamo of the shunt wound type, as shown by the accompanying plate.*



THEORETICAL DIAGRAM OF SHUNT WOUND DYNAMO CONNECTIONS.

Details of Tests taken March 22d, 1890.—

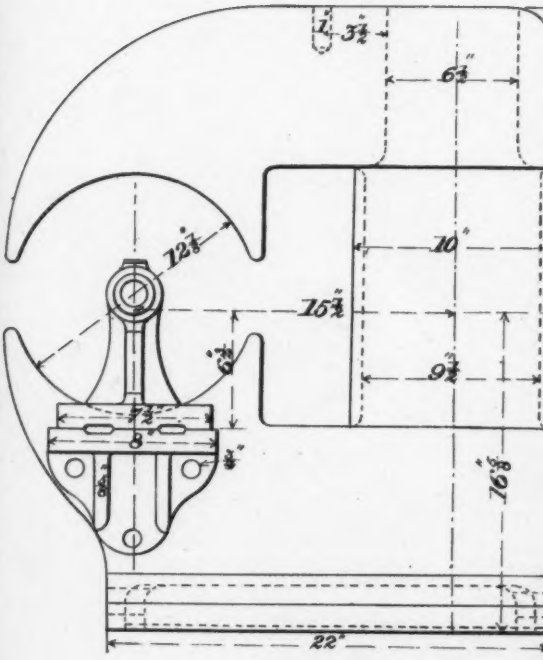
(See the accompanying plate for sizes of the dynamo.)

Resistance of armature,	R_a = .12 ohm.
Resistance of field magnets,	R_s = 33. ohms.
†Current in working circuit,	C_e = 50 amperes.
†Current in field magnet coil,	C_s = 3.85 amperes.
†Current in armature,	C_a = 53.85 amperes.

* This dynamo was tested by Prof. Jamieson on March 22d, 1890, at Messrs. McWhirter, Ferguson & Co.'s Works immediately after being made and before being delivered to the purchaser, Mr. Alexander Stephen, head partner of the Linthouse Shipbuilding and Engineering firm, Glasgow.

The result of his tests, the reduced drawings and his calculations have been inserted here as an appendix to the original paper read before the Institution of Engineers and Shipbuilders in Scotland, as likely to prove of special interest, not only on account of the recent date on which they were made, but also from the fact that the calculations have been done in absolute or centimeter, gramme, second (C.G.S.) units. This method is more definite and becoming generally adopted by electrical engineers.

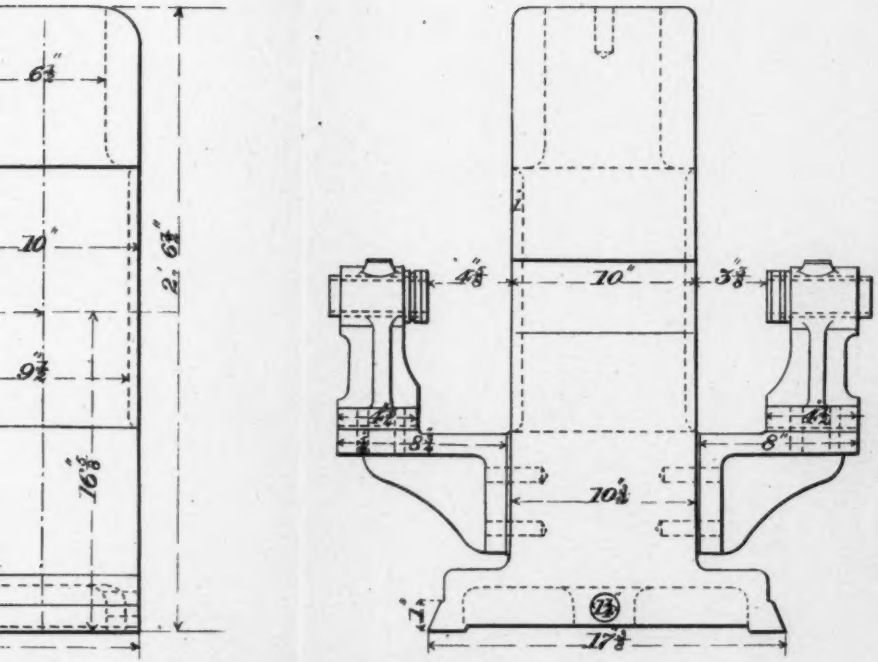
† The efficiencies were calculated as above from these particular results, but the following set of characteristic curves, drawn from the results obtained, give the corresponding currents and voltages throughout the full range of the tests.



7 UNIT

Scale

Tested at

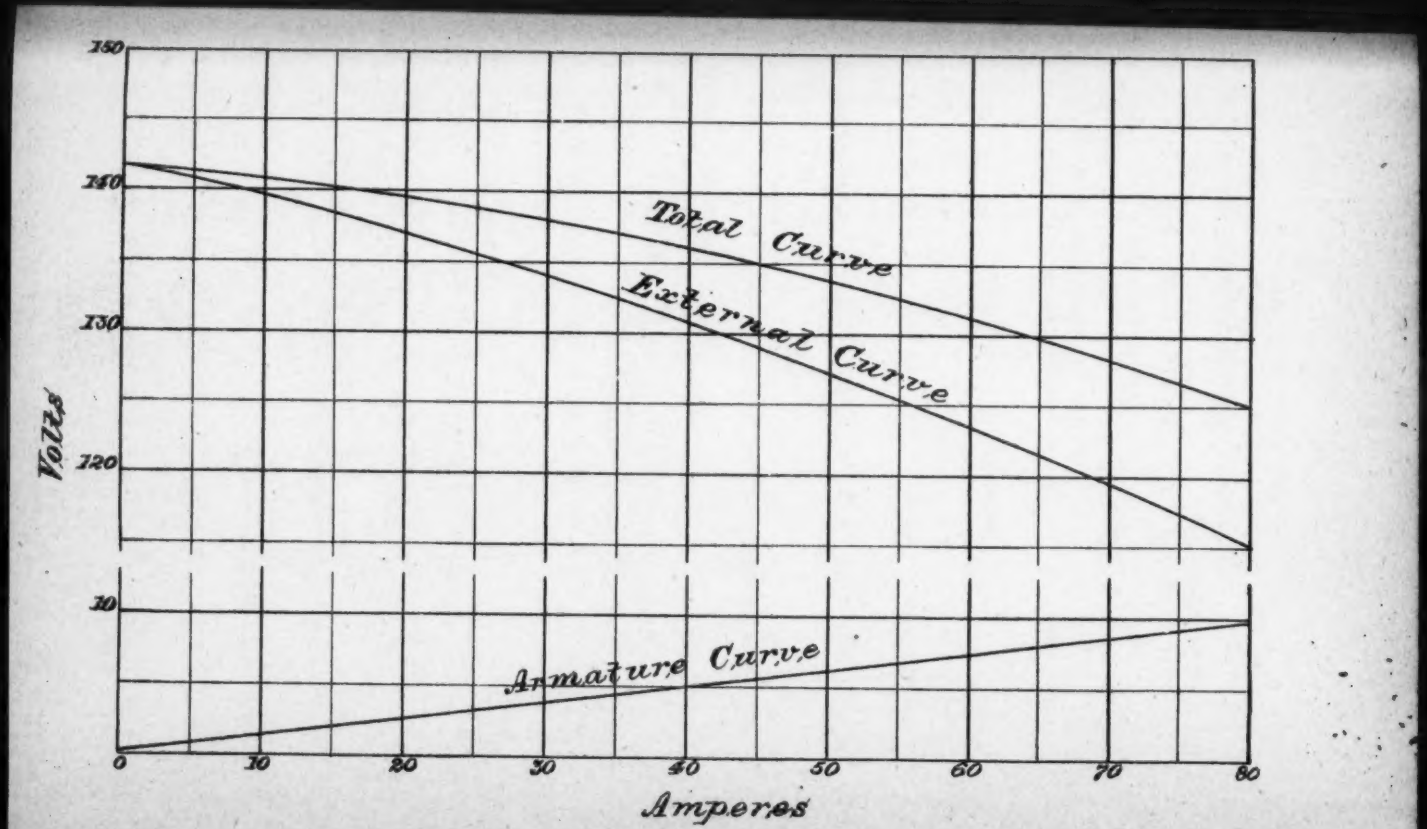


UNIT FARADAY DYNAMO

(Shunt wound)

Scale 0 3 6 9 12 ins.

Tested at Govan, March 22. 1890.


150
140
130
120
Volts
Total Curve
External Curve
10
Armature Curve
0
10
20
30
40
50
60
70
80
Amperes

Difference of potential at dynamo terminals,	$E_e = 127$ volts.
E.M.F. generated in armature,	$E_a = 134$ volts.
Number of commutator bars,	$= 36$
Turns of armature wire,	$T_{aw} = 288$
Armature revolutions per minute,	$N_{ar} = 850$
Number of turns in field magnet coil,	$= 4{,}136$

ELECTRICAL EFFICIENCIES.

External electrical power, or "output," available for electric lighting, etc., $= C_e \times E_e = 50 \times 127 = 6{,}350$ watts $= 88.47\%$

Power absorbed by armature, $= C_a^2 \times R_a = 53.85^2 \times .12 = 347$ watts $= 4.83\%$

Power absorbed shunt magnet coil, $= C_s \times E_e = 3.85 \times 127 = 488$ watts $= 6.7\%$

Total electrical power, $= (C_e \times E_e) + (C_a^2 \times R_a) + (C_s \times E_e) = 7{,}185$ watts.

From the measured dimension of the dynamo we found—

$L_m = 68$	inches	$= 172.7$	cm.
$L_{as} = .375$	"	$= .95$	cm.
$L_a = 13$	"	$= 33$	cm.
$S_m = 86.58$	sq. ins.	$= 558.6$	cm^2.
$S_{as} = 158.875$	"	$= 1025$	cm^2.
$S_a = 28.7$	"	$= 185.2$	cm^2.

Now since $E_a = 134 \times 10^8$ (C.G.S. units)
$N_{ar} = 850 \div 60$ per second.
$T_{aw} = 288$.

$$M_a = \frac{E_a}{T_{aw} \times N_{ar}} = \frac{134 \times 10^8 \times 60}{850 \times 288} = 3{,}285{,}000 \text{ C.G.S. lines.}$$

$$B_a = \frac{M_a}{S_a} = \frac{3{,}285{,}000}{185.2} = 17{,}730 \text{ C.G.S. lines.}$$

* By Hopkinson's table $\mu_a = 110$.

$$R_a = \frac{L_a}{S_a \times \mu_a} = \frac{33}{185.2 \times 110} = .00162$$

$$R_{as} = \frac{2\,L_{as}}{S_{as} \times \mu_{as}} = \frac{2 \times .95}{1025 \times 1.36} = .00136$$

Excitation for Armature.†

$$A \times T_{mw} = \frac{M_a \times R_a}{1.257} = \frac{3{,}285{,}000 \times .00162}{1.257} = 4233.$$

Excitation for Air Space.†

$$A \times T_{mw} = \frac{M_a \times R_{as}}{1.257} = \frac{3{,}285{,}000 \times .00136}{1.257} = 3562.$$

Assuming $\lambda = 1.25$

$$M_F = 1.25 \times 3{,}285{,}000 = 4{,}106{,}000$$

$$B_f = \frac{M_F}{S_{as}} = \frac{4{,}106{,}000}{558.6} = 7350$$

By table, $\mu_f = 125$

$$\therefore R_f = \frac{L_f}{S_f \times \mu_f} = \frac{172.7}{558.6 \times 125} = .002473$$

Excitation for Field Magnets.†

$$A \times T_{mw} = \frac{M_F \times R_f}{1.257} = \frac{4{,}106{,}000 \times .002473}{1.257} = 6463$$

Total ampere-turns $= 4333 + 3562 + 6463 = 14258$

Actual turns $= 15924$

The calculated ampere-turns are thus 10.5% less than the actual.

Assuming $\mu_f = 120$

$R_f = .002575$

$$\therefore A \times T_{mw} = \frac{M_F \times R_f}{1.257} = \frac{4{,}106{,}000 \times .002575}{1.257} = 9181$$

$\therefore$ Total excitation $= 16{,}976$.

The calculated ampere-turns are in this case 6.6% more than the actual.

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* See Munro & Jamieson's Pocket-book, 6th Edition, p. 380 *a*, for Hopkinson's table. Drawing a curve in accordance therewith we get $\mu_a = 110$.

† See Munro & Jamieson's Pocket Book, 6th Edition, p. 380.

It is evident when designing dynamos that a small difference in the assumption of the permeability of the field magnet cores will cause a great difference in the necessary excitation or ampere-turns. Nevertheless the calculation gives approximate values, and by applying the trial coil, as explained at p. 319, the exact number of ampere-turns required is found with a minimum of trouble.

XVII.

SPEED TRIALS OF FAST SHIPS.

By A. Denny, Esq., of the firm of Wm. Denny & Bros., Dumbarton, Scotland.

[Contributed in a letter to Assistant Engineer H. P. Norton, U. S. Navy, as a discussion of the article on page 68 of the current volume, and printed with the writer's sanction.]

Speaking now of the paper on speed trials of fast ships, I consider it a very valuable one.

I do not suppose any firm has had such a large experience in speed trials as mine. My late brother, Mr. Wm. Denny, first introduced the system of progressive trials, and we have made such trials of all the steamers built by us for very many years. I believe that you will agree with me, and I gather that you do so from your paper, that patent logs, which can be purchased as commercial articles, are totally useless for accurate speed trials.

To ascertain the correct speed of a ship, there is really only one way, and that is over an accurate measured mile, and by making runs in opposite directions under suitable circumstances, and on a properly conducted trial. The maximum speed, got from this kind of trial, must be clearly distinguished from the speed which can be obtained on a four or six hours run, which is really an endurance trial. The method No. 3, you suggest, is in my opinion, the only fair one for such a trial. Method No. 4, except under peculiarly favorable circumstances, would be unre-

liable, and this appears from the discussion to be the general opinion.

I think I cannot do better than give you a description of our method of trial. If you will look at the map of Scotland, and at the Firth of Clyde, you will see a place called Greenock; opposite this is the Tail of the Bank, the principle anchorage for inward and outward bound steamers. After the steamers leave our yard at Dumbarton we take them down to the Tail of the Bank, and on the day of the trial, which generally commences about 10 o'clock, steam is got up and everything is ready when we go on board, and we start immediately for the measured mile. If you look on the map, you will see a place called Wemyss Bay; south of that the coast line for some miles is almost straight, and it is there that the Skelmorlie measured knot is situated. This knot is certified by the Government as correct.

From the Tail of the Bank to Wemyss Bay we generally find is a sufficient distance for getting the fires in order and the steam up to its maximun pressure. From Greenock to Wemyss Bay we run and adjust the revolutions until it is found that the firemen can just keep up the steam constantly, the engines running steadily. We then put her on the mile, commencing generally at the north end and run down. After passing the lower posts, we run for about a mile and a half, and then turn as quickly as possible, straighten her up, and perhaps we run a mile steadily before we go on the measured mile again; run up the mile, and so on.

As a general rule for our own trials we commence by doing two double runs at the highest speed; we then, immediately on leaving the mile, reduce the revolutions by linking her up, keeping everything as steady as possible, and proceed to get a second pair of runs at a slightly lower speed, and so on for the other speeds. These are trials that we make for our own satisfaction, and for a scientific purpose, and we do our utmost to make them accurate.

As to taking times, etc., our method is as follows: We have six or seven chronographs (watch form), which we regulate as carefully as possible. There are generally three observers on

the bridge, and the engine-room staff below are also supplied with chronographs. One of the observers on the bridge, when nearing the mile, rings the telegraph "stand by," and when the posts come opposite one another—that is, when the ship is on the mile—he rings "full speed." This does not mean that the speed of the engines is increased, but simply informs the engine-room staff that the mile has begun. The observer at the counter reads it, and the revolutions are also checked by counting during the mile. Sets of diagrams are taken immediately on commencing the mile, and if there is sufficient time a set is taken half way, and another set just before leaving the mile. We have always plenty of trained assistants for this work.

On nearing the end of the mile the bridge observer rings "stand by," and when the posts come opposite he rings again "full speed." The observer at the counter reads it, and the revolutions are taken from this reading checked roughly by the observations during the run.

I sent you the progressive trials of the *Alfonso*, and if you plot the curve to any scale I think you will find the observed results lie upon a very fair curve.

It seems to me that both in your paper and in the discussion, your Society were endeavoring to devise methods of trying a ship which will give absolute scientific accuracy. Now, this is absolutely impossible. You speak of an error of a revolution which may creep in from reading the counter, and you also speak of reading the time to the fifth of a second. We never attempt to do it closer than half a second, and for this reason: We find that even on such a short period of time as three minutes, four observers will have a maximum difference of time, in some cases of a second. This is due to errors in the watch—that is to say, that some of the watches do not start promptly or stop promptly. It is also due to the personal error of the observer, which we have found by experiment to be so variable as not to admit of correction. It therefore appears to me useless to attempt such accuracy as the fifth of a second.

Now, there are other reasons which make it impossible to obtain anything like absolute accuracy, and the first is the question of the course.

a. The steersman may steer a serpentine course, which would naturally increase the length of the mile, and would also reduce the real speed of the steamer by the increased resistance when the rudder is put over.

b. The steersman may steer a course which is not at right angles to the posts. I may say that we check this when on the mile by the compass, but as a rule the error is so small as not to be worth correcting.

2d. There is the error arising from the variation of the tide. This should not be great if the trials are run promptly. Fortunately for us, owing to the sheltered position of our mile, the tide is small, but, on the other hand, it is very irregular and does not follow any law. The Admiralty, in their trials, endeavor to eliminate this error by taking what is called the Admiralty mean, but I very much question whether this is a wise thing to do.

3d. There is the error due to wind; for absolute scientific accuracy there should be absolutely none. This is a condition which, it is needless to say, never really occurs.

We have made experiments in our tank to ascertain the effect of the air resistance of the rigging and masts, and we found it very great indeed in models. The chances are it would not be so great, proportionately, in a full-sized ship, but there are no data for this.

Taking, then, all the chances of error, we find as the result of many years' experience that our method of making trials, which I have described, is of sufficient accuracy even for scientific purposes.

There is another method of making trials which is in vogue here. If you look again at the map you will see that the Firth of Clyde is peculiarly well adapted for trials, being very much sheltered. You will see a lighthouse called the Cloch, near to Greenock. Further down the Firth you will see two islands called the Greater and Lesser Cumbrae. On the Lesser Cumbrae is situated the Cumbrae Light; the distance between the Cloch and Cumbrae Lights is 13.666 knots, and if there are no winds and the trial is commenced about the middle of the ebb, with a ship steaming about 21 knots, the tide is practically eliminated by performing two double runs, and this is what we did in

the case of the *Henriette*. The result obtained, 21.1 knots, when taken in conjunction with the power developed, agreed almost exactly with our measured mile trials. Long turns were made at each end which were not counted in the speed, the speed being taken only between the two lighthouses. As we ran for practically three hours there was not the slightest chance, nor was there any need, for bottling. The trials were watched, on behalf of the Belgian Government, by no less than seven officers. Unless, however, these trials are made in good faith, under favorable conditions of wind, and at such a time of the tide as I have mentioned, it is quite possible to produce a result which may be half a knot or even three-quarters of a knot in excess of the real speed.

As there is only one point on shore, two points are taken on the ship, one at each side, but with a skilful steersman, he has only to turn in toward the Cloch on starting, and out from the Cumbrae on ending, to shorten the time, as you will easily see, very considerably. This, combined with the tide and wind reasons, is my objection to your fourth method.

As I said before, in my opinion, your third method is the correct one, but I am afraid you must give up all idea of such absolute accuracy as you seem to hope for. At the same time, I think it would be advisable to have a rotary revolution counter with long index hands, such as suggested by one of the gentlemen who spoke. There will be no harm in getting the revolutions as accurately as possible. You should also have watches most carefully rated, and have numerous observers who can be relied upon to act with perfect fairness; read their watches as closely as possible even to the fifth of a second, and take the mean. I do not believe by any method that you will arrive at a more accurate result than you will have thus obtained.

I think your standard for wind is too high, and where, as in a Government ship, there is, as a rule, no great hurry for her trials, I would certainly wait for a calm day.

In the case of these Belgian boats we had £500 per tenth of a knot premium or penalty, and the trials were to be performed in fine weather. Our guarantee was 20½, we knew we could do at least 21.1, and we did one day on two runs 21.28, when the

trials had to be stopped on account of fog. The day that the trials were completed we had a 16-knot breeze against us, this when added to 21 knots gave a 37-knot breeze against us going down, and a 5-knot breeze against us coming up, which spoiled our result to a certain extent.

In the case of these steamers the Government insisted that the premium and penalty should only be paid or exacted for each completed tenth of a knot. The official speed of the *Henriette* was 21.0925 and we lost £500 premium by .0075 of a knot. I think, however, that premium should be paid or exacted at the rate of, not for completed, tenths.

I may inform you that we have tried Mr. Biles' method of taking speeds very often, but it is not reliable—for a scientific purpose at least.

Referring to the discussion, Mr. Emery seems to think that at the end of the mile when turning with everything open, bottling might occur. I can assure him from practical experience this does not occur to any extent worth noticing. Mr. Emery is, I think, wrong in the speed of the *City of Paris;* I think her fastest run was 20 knots per hour.

Mr. Weaver refers to our experimental tank propeller-logs. In using these logs we have them a very considerable distance ahead of the steamer. They are most delicately made, and all they have to do is to make an electric contact underneath the water. We find by trial that to use such logs on board a ship is quite out of the question, as they cannot be fixed sufficiently far from the steamer to be uninfluenced by the current she sets up in passing through the water. These logs also require to be rated and checked every time they are used.

His suggestion as to a revolution counter is, I think, good, but after all so long as you have no automatic means for starting the counter on entering the mile, and stopping it on leaving, you have the personal error coming in which I have mentioned before. As I said, however, there will be no harm in getting the revolutions as closely as possible by this means.

We do not find in practice that anything like two miles is required for getting up a uniform speed. This refers to Mr. Weaver's remarks on the *Lepanto.*

Mr. Mattice wisely points out the defects of propeller recorders. I agree with him that the wind force of four is too great, but I would be sorry to sail on any boat whose propeller blades had taken a permanent set after a trial.

I should like to see the system of trials with the Emery testing machine tried before accepting it. As a contractor for official trials, I may say that, with my present knowledge, I have grave doubts of its accuracy. Referring to Mr. Warburton's remarks, I see no reason why the contractors should not run on the measured mile at their maximum speed. We always do so.

Mr. Dickie: The runs at the higher speeds should be closer than at the lower. Suppose a steamer was designed for 19 knots, then I would try to have a pair of runs at 19, at 18½, at 18, at 17, at 14, and about 11. You will see the reason for this.

It is a dangerous thing to produce a speed curve much beyond the highest speed attained, because humps may occur which would vitiate the result. See Mr. Froude's investigation.

Referring to Mr. Baird's remarks, I think they are very good. In regard to the steamboatmen's story: When I was in America I happened to ask about the speed of a small boat I saw, which I reckoned should be about 13 knots; I was told she had done 18. I inquired where, on the measured mile, and they said, No; on the Delaware River, between two known points. I then asked if they had made two runs, one with and one against the tide. My informant said, No; but now I mentioned it, he did think it was with the tide. I hope you won't think me rude if I say that, on this side up till now, we have always accepted American merchant speeds with a grain of salt.

I am glad that you are drawing attention to this important subject, and hope that the effect of your efforts will be to inaugurate both for the Navy and for the merchant service a correct system of measured mile trials.

I have written this letter from day to day, as I found time, hence the delay in answering your last kind letter.

With kind regards, believe me, yours sincerely,

A. DENNY.

XVIII.

REPORT UPON TRIALS OF THREE STEAMERS, *FUSI YAMA, COLCHESTER, TARTAR.*

By Prof. Alexander B. W. Kennedy, F. R. S., *Chairman.*

[Reprint of report of the Research Committee on Marine Engine Trials, read before the Institution of Mechanical Engineers.]

I.—TRIAL OF THE S.S. *Fusi Yama.*

Steamer.—This steamer was tested as a good example of an ordinary trading vessel working under the usual conditions. She belongs to Messrs. Gellatly, Hankey, Sewell & Co., to whom the committee are much indebted for giving them every facility for making the experiments. She is a vessel of 214.3 ft. in length between perpendiculars, 29.3 ft. beam and 20.5 ft. depth. Her registered tonnage is 632 net, and her gross tonnage, 994; under deck, 899 tons. Her displacement on the day of the trial, when the mean draught was 18 ft. 11¾ in., was 2,175 tons.

The trial was made upon the 14th and 15th November, 1888, on a run from Gravesend to Portland. The ship left Gravesend, where her compasses had been adjusted, about six o'clock on the afternoon of the 14th November, and the trial started at 8.51 P.M. on that evening. It lasted until 11 A.M. on the 15th. The run was continuous, with the exception of stoppages amounting in all to 12 minutes between 3.10 and 3.25 A.M., when landing the pilot off Dover. In addition to this the engines were running slow for about 13 minutes between 3 and 3.30 A.M. The reading of the counter was entered at each change of speed or stoppage during this time.

Engines.—The *Fusi Yama* is fitted with compound surface-condensing engines, made by Mr. Martin Samuelson, of Hull, in the year 1874. They had been thoroughly overhauled by Messrs.

Rait & Gardener, under the superintendence of Mr. Frederick Edwards, immediately before the trial, and were in thoroughly good working order. The cylinders' diameters are 27.35 in. and 50.3 in., measured from gauges. Their common stroke is 33 in. The diameter of both piston rods is 4.9 in. There are no tail rods.

The cranks are at right angles, the low pressure leading.

The cylinders are not jacketed. The clearances of the high and low pressure cylinders, as measured on the drawings, are respectively 7.6 and 5.0 per cent.

Each cylinder is provided with a single slide-valve worked by the ordinary link-motion gear. The throttle and stop valves and the link-motion were sealed up at starting. The link-motion, however, was opened out a little at 1.24 A. M., and remained in its new position during the rest of the run. The screw propeller is four-bladed, having a diameter of 13 ft. and a mean pitch of 16.37 ft.

Boilers.—Steam is supplied by one boiler 13 ft. 3¾ in. mean diameter, 11 ft. long, and containing three furnaces. The total grate surface is 52 square feet, and the total heating surface 2,257 square feet, the ratio between the two being 1 to 43.4. The mean diameter of the flues is 2 ft. 11½ in. The fire-bars are of the ordinary description, and there are 17 bars in the width of each furnace, and therefore 34 in each grate. There are 232 tubes of 3¾ in. external diameter and 7 ft. 5 in. long between plates, equivalent to a total tube surface of 1,689 square feet, which is 32.5 times the grate surface. The internal diameter of the funnel is 4 ft. 6½ in., and its total height is 43 ft. 2 in. above the center of the lowest furnace.

The total weight of the engines and boiler, including water in condenser, pipes and boiler, and all mountings, is about 100 tons. The net volume of the boiler is about 1,681 cubic feet.

Duration of Trial.—The duration of the trial from start to finish was 14 hours 9 minutes. Deducting the 12 minutes of stoppage, the running time was therefore 13 hours 57 minutes, or 837 minutes.

Coal Measurement.—The same method was used for weighing the coal as was employed upon the *Meteor* trial. About 70 lbs. of coal were weighed in each bucket, and about nine buckets were weighed on to the stoke-hold floor at one time. The trial was started with a clean floor. The time was noted at which each lot of weighed coal was put upon the fire, and no more coal was weighed out until the floor was again clear. The fires were not cleaned during the run. The ashes were weighed after the trial was over. The coal was West Hartley Tyne coal, costing 15s. 9d. per ton. It was somewhat irregular in quality, but a sample collected at different times during the run, and afterwards well mixed and pounded, gives the following analysis, which has been made by Mr. C. J. Wilson:

	Coal as used.	*Dry Coal.*
Carbon	70.85 per cent.	77.52 per cent.
Hydrogen	4.71 "	5.15 "
Moisture	8.60 "	0.00 "
Ash	5.11 "	5.59 "
Nitrogen, Sulphur, Oxygen, &c., by difference	10.73 "	11.74 "
	100.00	100.00

The calculated caloric value of this fuel as used is 12,760 thermal units per lb., which is equivalent to the evaporation of 13.21 lbs. of water from and at 212° Fahr. The equivalent carbon-value of this fuel as used is 0.878 lbs. per lb. The total coal used was 13,768 lbs. in 849 minutes. As, out of the whole time of running, the engines were stopped for 12 minutes, during which period, of course, the fires were more or less damped, it may be taken that the actual time of firing was equivalent to about 840 minutes; but in order not to confuse matters the boiler time will be taken as equal to the engine time, that is to say, to the time during which the engines were actually running, namely 837 minutes. The total coal was therefore 16.45 lbs. per minute, or 987 lbs. per hour. The weather was fair until reaching Dover, but became very rough during the last part of the trial, which was brought to an end somewhat sooner than was intended, because of the great difficulty of obtaining further accurate measurements. Up to the actual point of ceasing observations,

however, the weather did not affect their accuracy, although it rendered difficult the estimation of the water-level in the boiler at the end of the trial. The roughness of the weather at the end of the trial prevented accurate measurements being made as to the amount of clinker obtained from cleaning the fires. The actual amount of ash weighed off from under the grates was only 278 lbs., or about 2 per cent. of the total weight of the fuel.

Furnace Gases.—The temperature of the chimney gases was observed 15 times during the trial, and its average value was 578° Fahr.; the highest reading being 617°, and the lowest (which was just after starting) 500°. Two samples of furnace gases were collected and analyzed; the analyses of these by weight and by volume are given in the following table:—

PERCENTAGE COMPOSITION OF FLUE GASES.

Constituent.	No. 1.		No. 2.		Mean.	
	By Volume.	By Weight.	By Volume.	By Weight.	By Volume.	By Weight.
Carbonic Acid	6.93	10.30	8.14	12.04	7.53	11.17
Carbonic Oxide	0.00	0.00	0.00	0.00	0.00	0.00
Oxygen	12.14	13.13	11.00	11.84	11.57	12.48
Nitrogen	80.93	76.57	80.86	76.12	80.90	76.35
	100.00	100.00	100.00	100.00	100.00	100.00

The mean chimney draft, which was measured by a U gauge-glass at the place where the furnace gases were collected, was 0.28 inch of water.

Feed-water Measurement.—The feed-water was measured in the same way as for the *Meteor*, on its way from the hot-well to the feed-pump of the engine. The tanks used for weighing it (which held about 260 lbs. each) were placed on the upper engine-room platform; their contents were carefully tested by weighing water into them and they were at the same time ascertained to be free from any leakage. The necessary correction having been made for the temperature of the feed, the total water used amounted

to 109,800 lbs. (its mean temperature being 129.5° Fahr.), over a total time of 850 minutes. The engine was actually running during 838* minutes of this time, so that the mean consumption of water per minute (engine time) was 131.0 lbs., or 7,860 lbs. per hour.

All the steam made by the boilers went to the main engine, steam being kept up in the donkey boiler for any other purposes that were required.

Power Measurement.—Indicator diagrams were taken at about half-hourly intervals throughout the whole of the trial, 26 sets or 104 cards being taken in all. A pair of Darke's indicators were used on the high-pressure cylinder, with $\frac{1}{80}$th springs, and a pair of Richards' indicators on the low-pressure cylinder, with $\frac{1}{10}$th springs. The following are the mean effective-pressures in the two cylinders in pounds per square inch:—

Cylinder.	*Top.*	*Bottom.*	*Mean.*
High-pressure	31.06	30.41	30.74
Low-pressure	10.43	11.30	10.87

These pressures correspond with the following I.H.P.'s:—

High-pressure cylinder	168.2
Low-pressure	203.1
Total indicated horse-power	371.3

One set of cards was taken during the time when the engines were slowed down (13 minutes), and the proper allowance has been made for this in the preceding averages.

The maximum I.H.P. given by any one set of cards was 402.2, which occurred at 5:30 A. M., with 57.5 revolutions per minute, and 59.0 lbs. of steam. The minimum I.H.P. by any one set of diagrams at full speed was 313.0, at 11:00 P. M., with 53 revolutions per minute, and 49 lbs. of steam. From the 104 diagrams which were taken during the trial a mean diagram has been plotted for each of the two cylinders.

*The last feed-tank was emptied at 11:1 A. M., or one minute after the last reading of the counter.

Speed.—The counter read 8,507 at 8:51 P. M., when the trial commenced, and 55,033 at 11:00 A. M., when the trial ended. The total number of revolutions made by the engines was therefore 46,526 in a total running time 837 minutes, which gives an average rate of 55.59 revolutions per minute. The maximum number of revolutions per minute for any half hour was 57.5, and the minimum number (except when slowed down) was 53.0.

Pressures, &c.—The mean barometric pressure during the trial was 30.1 in. of mercury, or 14.8 lbs. per square inch. The mean boiler pressure was 56.84 lbs. per square inch above the atmosphere, measured on the gauge in the stoke hold. On the engine-room gauge the mean pressure was 55.2 lbs. to the square inch, and the pressure was fairly constant, except during about two hours towards the beginning of the run, and during the time when the engines were slowed down. The pressure-gauge in the stoke hold was read every quarter of an hour during the trial, and the other gauges were read every half hour alternately with the counter reading, and at the same time as that at which the indicator cards were taken. The mean reading of the vacuum gauge was 25.40 inch of mercury, or 12.48 lbs. per square inch below the atmosphere. The mean initial pressure in the high-pressure cylinder (from measurement of diagrams) was 50.3 lbs. per square inch above the atmosphere; and the mean back-pressure in the low-pressure cylinder was 10.9 lbs. per square inch below the atmosphere, which corresponds to an absolute back-pressure of 3.9 lbs. in the cylinder.

Boiler Efficiencies.—The rate of combustion in the furnaces was 18.98 lbs. of coal per square foot of grate surface per hour, or 0.437 lb. per square foot of total heating surface per hour. The evaporation was at the rate of 7.96 lbs. of water per lb. of coal. As the feed entered the boiler at a temperature of 129.5° Fahr., and as the temperature corresponding to the mean steam pressure was 304° Fahr., each lb. of steam must have taken up 1,077 thermal units; so that the equivalent evaporation from and at 212° Fahr. was 8.87 lbs. of water per lb. of coal, or about 10.10 lbs. per lb. of carbon-value in fuel. The equivalent amount of heat utilized per lb. of coal was 8,570 thermal units, or say

67.2 per cent. of the whole calorific value of the fuel; which percentage therefore represents the actual boiler efficiency.

The total nominal calorific value of the fuel burnt per minute was 209,900 thermal units. Assuming that the two samples of furnace gases analyzed gave a fair average value, it appears that the weight of dry air per lb. of coal was about 22.8 lbs. The loss of heat in raising the temperature of the furnace gases from 55° Fahr., allowing for the steam due to the combustion of the hydrogen in the fuel, works out to 2,995 thermal units, or 23.5 per cent. of the whole calorific value of the fuel. There was no loss by formation of carbonic oxide. The loss due to evaporation of the moisture in the fuel would be 0.9 per cent. These quantities add up to 91.6 per cent. of the whole heat of combustion, the balance of 8.4 per cent. including all losses by radiation. The weight of water evaporated per square foot of total heating surface was 3.48 lbs. per hour. The average rate of transmission of heat through the material of the boiler was 3,750 thermal units per square foot of total heating surface per hour.

Coal Consumption.—The total coal burnt, namely, 987 lbs. per hour, corresponds to 2.66 lbs. per I. H. P. per hour. This is equivalent to 2.33 lbs. of carbon value per I. H. P. per hour.

Engine Efficiencies.—The measurement of feed water shows that the quantity used per I. H. P. per hour was 21.17 lbs. per hour. The actual heat received by the feed water per minute was 141,100 thermal units, or 380 thermal units per I. H. P. per minute, which is 67.2 per cent. of the whole heat of combustion. The absolute engine efficiency, or ratio of the heat turned into work to the total heat received by the feed water, was 11.25 per cent.

Total Efficiency.—The combined efficiency of the boiler and engines, or the ratio of the heat turned into work to the total heat of combustion of the fuel, was 0.672×0.1125, which is equivalent to 7.6 per cent.

Steam from Indicator Diagrams.—The following are the results of measurements made upon all the indicator diagrams taken to ascertain the proportion of steam accounted for by them.

The actual weight of feed water used per revolution was 2.357 lbs.:

Proportion of steam accounted for by indicator diagrams.	Lbs. per revolution.	Percentage of total feed.	Percentage present in cylinder as water.
Steam present in high-pressure cylinder after cut-off, when the pressure was 45.2 lbs. per square inch above the atmosphere	*Lbs.* 2.08	*Per cent.* 88.2	*Per cent.* 11.8
Steam present in low-pressure cylinder, near end of expansion, when the pressure was 4.8 lbs. per square inch below the atmosphere	1.66	70.7	29.3

2.—TRIAL OF THE S.S. *Colchester.*

Steamer.—This vessel is the latest of those built for the Great Eastern Railway Co., for carrying their passenger traffic between Harwich and Antwerp. The committee is very greatly indebted to the Great Eastern Railway Co., and especially to Mr. Holden, the locomotive superintendent of the line, and to Captain Howard, the company's marine superintendent, for the facilities which they gave throughout for the carrying on of the trial. The *Colchester*, which is driven by twin screws, is a vessel of 281 ft. in length, 31 ft. beam, and 15.2 ft. in depth. Her gross registered tonnage is 1,160, her net registered tonnage being only 517 tons, the remainder of 643 tons representing space occupied by machinery, seamen, &c. Her draught at Harwich at the end of the trial was 13 ft. 7 in. aft and 10 ft. 6 in. forward, corresponding to a displacement of 1,675 tons.

The trial was made upon the 9th November, 1889, on a run from the Humber to Harwich. The ship had been lying in the Humber, where her machinery had received a thorough overhauling by the makers, Messrs. Earle's Shipbuilding and Engineering Co., Hull. Steam was got up on the morning of the 8th November, and the engines were turned round about 3 P. M. on that day. The steamer left her moorings about 3 A. M. on the

9th November, and the trial started at 5 A. M. on that morning. It lasted until 4 P. M. exactly, the boat being taken a little south of the Stour estuary in order to prolong the trial somewhat. The furthest south point was reached at 3 P. M. on 9th November. The engines were stopped about seven minutes during this time to allow the pilot to leave the ship, and they were also slowed down for five minutes about the same time. The counter was read at each of these changes of speed, all of which occurred before 5.40 A. M.

Engines.—The *Colchester* is fitted with two compound surface-condensing engines, driving the two screw shafts. The engines are entirely separate, and are placed slightly inclining from the vertical, the space between them being taken up by a condenser which is common to the two. They were built by Earle's Shipbuilding and Engineering Co. in 1888–'89. The vessel had been on her station running to Antwerp and back twice every week from the end of February to the middle of September, 1889, and had run a total distance of about 16,530 miles during that time; and had undergone her first overhaul at the makers', as already mentioned, during the week before the trial. The engines and boilers therefore were in thoroughly good order. The diameters of the cylinders are 30 in. and 57 in. respectively. An opportunity has not occurred to gauge them. Their common stroke is 36 in. The diameter of all piston rods is 6.0 inches, and of all tail rods 4.5 inches. The cranks of each engine are at right angles, and the high-pressure cranks lead. The cylinders are not jacketed. The clearances of the high and low-pressure cylinders as given by the owners are 9.39 and 6.23 per cent.

The valve gear is of the ordinary link motion type. The surface-condenser contains 3,000 square feet of tube surface in 1,176 tubes, 1 in. external diameter, and 9 ft. 11¾ in. long over plates. The screw propellers have cast-steel blades and boss; each is 11 ft. 6 in. diameter, and 20 ft. 4½ in. pitch, with a surface of 45½ square feet.

Boilers.—There are two iron boilers, each double-ended, 13 ft. mean diameter, and 18 ft. 3 in. long; each boiler having six furnaces of 3 ft. 4 in. internal diameter. The total grate surface

is 220 square feet; the length of each surface on bars being 5 ft. 6 in. The total heating surface is 5,820 square feet, of which 4,770 square feet is tube surface. There are 784 tubes, 3½ in. external diameter, and 6 ft. 9 in. long over plates. The total heating surface is therefore 26.5 times, and the tube surface 21.7 times the grate surface. The boilers have been designed to work safely with a pressure of 90 lbs. of steam, but are ordinarily worked, as during the trial, with a pressure from 5 to 10 lbs. less. There are two double funnels, one for each stoke-hold; the internal diameters of their outer and inner shells are respectively 6 ft. 7 in., and 5 ft. 5 in., and the total height of each funnel is 47 ft. above the center of the wing furnaces.

The total weight of the engines and boilers, including water in condenser, pipes and boilers, and all mountings, is about 395 tons. The net volume of the boilers is about 4,980 cubic feet.

Duration of Trial.—The duration of trial from start to finish was 11 hours exactly. Deducting the 7 minutes of stop, the running time was therefore 10 hours 53 minutes, or 653 minutes.

Coal Measurement.—The coal was weighed in each stoke-hold by spring balances as before. From 500 to 700 lbs. of coal was put down at one time on each side of each stoke-hold. The trial was started with clean floors in each stoke-hold. The time was noted at which each lot of weighed coal was put upon the fire, and no more coal was weighed out until the floor was again clear. The fires were not cleaned during the run. The ashes were weighed after the trial was over; but the weight of the ashes given below includes the ash formed while steam was being raised during the time the fires were banked up before the start, and during the two hours' run before the trial actually commenced. The fire-grates were fitted with Henderson's fire-bars, with the object mainly of rendering possible a somewhat large rate of combustion, so as to get the greatest possible duty out of the boilers. The coal used in the forward stoke-hold, and during the first half of the run in the after stoke-hold, was Monk Bretton (Yorkshire) coal. For the second half of the run a mixed coal, chiefly Hucknall and Shireoaks (Nottinghamshire), was used in the after stoke-hold. Coal samples were taken

frequently during the run from both stoke-holds, and mixed together, and finally pounded; so as to obtain as representative a sample as possible for analysis. The analysis was made as before by Mr. C. J. Wilson, who gives it as follows:—

	Coal as used.	*Dry Coal.*
Carbon	71.89 per cent.	75.08 per cent.
Hydrogen	5.42 " "	5.66 " "
Moisture	4.25 " "	0.00 " "
Ash	4.08 " "	4.26 " "
Nitrogen, Sulphur, Oxygen, &c., by difference	14.36 " "	15.00 " "
	100.00	100.00

The following is an analysis made by Mr. Wilson of a sample of the clinker and ash:—

Loss on ignition (= Carbon)	43.73 per cent.
Mineral matter	56.27 " "
	100 00

The ash therefore contained about 43.7 per cent. of carbon.

The calculated calorific value of the fuel as used is 13,280 thermal units, which corresponds to the evaporation of 13.75 lbs. of water from and at 212° Fahr. The equivalent carbon-value of 1 lb. of the fuel as used is 0.913 lbs. The total coal used was as follows:—

After	stoke-hold,	port side	16,020 lbs.
"	"	starboard side	15,426 "
Forward	"	port side	15,509 "
"	"	starboard side	15,536 "
			62,491 lbs.

This amounts to 95.7 lbs. per minute, or 5,742 lbs. per hour, taking the running time of 653 minutes. As a check upon the coal consumption it has also been worked out from a time after the stoppage, at about 5:40 A. M., to the end of the trial; and over this period it comes to 95.9 lbs. per minute, that is to say, the same as before within the limits of measurements.

The weather was very fine during the whole of the trial, but a little foggy towards the end; which made the captain of the steamer reluctant to continue the run longer, as it was originally intended to do.

The actual amount of clinker and ash weighed off under the conditions already mentioned, was 2,890 lbs. It may be taken roughly that of this quantity, 2,200 lbs., or 200 lbs. per hour, was made during the actual run, which is about 3½ per cent. of the total quantity of coal.

Furnace Gases.—A sample of the furnace gases was collected over mercury by Mr. Wilson each hour from each funnel, there being 20 samples in all. The analyses of these samples are given in Table 3. This set of analyses and the corresponding set for the *Tartar* trial, are perhaps the most important and complete which have ever been made under such conditions, and merit very close examination. The "mean" figures in the table show that 3.8 per cent. by weight of the dry furnace gases is carbon, from which it follows that, allowing for the combustion of the hydrogen of the fuel, 18.5 lbs. of dry air passed through the furnace per pound of coal burnt.

TABLE 3.

"COLCHESTER" TRIAL.

Analyses of Funnel Gases by Volume.

	No. of Sample.	Carbonic Acid.	Carbonic Oxide.	Oxygen.	Nitrogen.
		Per cent.	Per cent.	Per cent.	Per cent.
FORWARD FUNNEL.	1	8.13	0.00	11.20	80.67
	3	10.55	1.84	6.59	81.02
	5	12.47	0.00	6.43	81.10
	7	7.98	0.00	11.79	80.23
	9	12.37	1.64	5.39	80.60
	11	9.89	0.00	9.72	80.39
	13	11.18	0.00	8.30	80.52
	15	10.60	0.00	9.17	80.23
	17	11.80	0.00	6.45	81.75
	19	11.27	0.00	7.67	81.06
	Mean.	10.62	0.35	8.27	80.76
AFTER FUNNEL.	2	7.75	0.00	11.27	80.98
	4	8.93	0.00	10.66	80.41
	6	8.72	0.00	11.21	80.07
	8	6.18	0.00	13.46	80.36
	10	6.33	0.00	13.69	79.98
	12	3.01	0.00	17.66	79.33
	14	9.97	0.00	9.72	80.31
	16	12.08	1.15	5.90	80.87
	18	9.90	0.00	10.37	79.73
	20	5.94	0.00	14.09	79.97
	Mean.	7.88	0.12	11.80	80.20
Mean of Both Funnels.		9.25	0.23	10.04	80.48

"COLCHESTER" TRIAL.

Analyses of Funnel Gases by Weight.

No. of Sample.	Carbonic Acid.	Carbonic Oxide.	Oxygen.	Nitrogen.	Time of Collecting.
	Per cent.	Per cent.	Per cent.	Per cent.	
1	12 02	0.00	12.05	75.93	5.20 a. m.
3	15 50	1.72	7.04	75.74	6.20 "
5	18.14	0.00	6.80	75.06	7 20 "
7	11.80	0.00	12.68	75.52	8.25 "
9	18.03	1.52	5.71	74.74	9.40 "
11	14 52	0.00	10 38	75.10	10.40 "
13	16.33	0.00	8.82	74.85	11.40 "
15	15.52	0.00	9.76	74.72	12.40 p. m.
17	17.22	0.00	6.85	75 93	2.5 "
19	16.47	0.00	8.15	75.38	3.5 "
Mean.	15.56	0.32	8.82	75.30	
2	11.48	0.00	12.15	76.37	5.50 a. m.
4	13.16	0.00	11.43	75.41	6.50 "
6	12 86	0 00	12.02	75 12	7.55 "
8	9.21	0.00	14.59	76.20	9.10 "
10	9 42	0.00	14.82	75.76	10.10 "
12	4.54	0.00	19.36	76.10	11.10 "
14	14.63	0.00	10.37	75.00	12.10 "
16	17.62	1.07	6.26	75.05	1.35 "
18	14.52	0 00	11.06	74.42	2.35 "
20	8.85	0.00	15.28	75.87	3.35 "
Mean.	11.63	0.11	12.73	75.53	
Mean of Both Funnels.	13.59	0.22	10.78	75.41	

The chimney temperatures were read in each funnel every quarter of an hour during the whole of the trial, from a mercury thermometer containing compressed nitrogen over the mercury. This thermometer reads up to 860° Fahr., but in a very large number of cases (about $\frac{3}{8}$ths of the whole) the temperature rose somewhat above the maximum reading of the thermometer. It was obvious from the motion of the mercury that the true temperature was not much above the limit of the thermometer, but of course its exact value is unknown. Assuming that the mean value of these readings was 20° Fahr. above the limit of the thermometer, or 880° Fahr. (which is probably true within an error of 10° either way), the mean temperature in the forward funnel was 828° Fahr., and in the after funnel 842° Fahr., the mean of both being therefore 835° Fahr.

The mean chimney draft measured by U gauges was 0.38. in of water in the forward, and 0.34 in. in the after funnel.

Feed-Water Measurement.—The feed-water was measured in two tanks. These tanks have been made specially for feed measurement, in a form as adaptable as possible to the conditions of various steamers; and in parts, of such a size that they can be readily got down into the engine-room of a steamer, and erected and jointed there. It was found possible to put them together in an extremely confined space in the *Colchester*, and again in the *Tartar* at a somewhat later date. Each tank held about 2,200 lbs. of water at 64° Fahr., between the upper and the lower cocks. Their contents were carefully tested by weighing water into them when jointed and connected, before placing them on board the steamer. Each tank lasted in the *Colchester* about three minutes. The total amount of water used was 468,845 lbs., over a total running time of 654 minutes.* This amounts to 717 lbs. per minute, or 43,020 lbs. per hour, the mean temperature of the feed being 113° Fahr. The feed-pump of one of the engines only was employed to pump throughout, the other pump not being used. All the steam made by the boilers went to the main engines; the steam for the circulating pump engine, the steering engine, and dynamo engine, and any

* The last feed tank was finished at 4:1 P. M. instead of 4 P. M.

steam for heating that was required, was taken from the donkey boiler, to which special steam connections were made for the purpose. The water-level was practically the same at the beginning and end of the trial.

Power Measurement.—Indicator diagrams were taken every half-hour from each engine. There were thus 22 sets of diagrams, or 176 single cards, taken in all. The six indicators used were made by Messrs. T. S. McInnes and Cairns, of Glasgow, and kindly lent by them to the Committee. These indicators are very simple in construction and handy in use, and their vulcanite sheathing is attended, at sea especially, with much convenience in manipulation. The following are the mean effective pressures in the cylinders in pounds per square inch:—

Cylinder.	*Top.*	*Bottom.*	*Mean.*
Port—high-pressure	43.34	47.96	45.65
" low-pressure	13.25	13.58	13.42
Starboard—high-pressure	38.45	45.68	42.07
" low-pressure	11.68	13.15	12.42

These pressures correspond to the following I.H.P.'s:—

Port engine—high-pressure cylinder	490 3	
" " low-pressure cylinder	532.2	
		1,022.5
Starboard engine—high-pressure cylinder	457.9	
" " low-pressure cylinder	499.3	
		957.2
Total indicated horse-power		1,979.7

The maximum I.H.P. given by any one set of cards is 2,194.2, which occurred at 10:15 A. M., with a mean speed of 88.6 revolutions per minute, and 86 lbs. of steam. The minimum I.H.P. by any one set of cards was 1,699.6, at 5:45 P. M., with a mean speed of 83.2 revolutions per minute, and 73 lbs. of steam. From the 176 diagrams which were taken during the trial, two mean diagrams have been plotted, one for the two high-pressure and one for the two low-pressure cylinders.

Speed.—The counters for the two engines were entirely separate. The total number of revolutions made by the port engine was 56,136, and the total number of revolutions made by the starboard engine was 56,857; which totals correspond respectively to 86.0 and 87.1 revolutions per minute. The maximum number of revolutions per minute for any half-hour was 89.9 for the port, and 91.7 for the starboard engine, and the minimum number, except when slowed down, was 82.9 for the port, and 84.0 for the starboard engine.

Pressures, &c.—The mean barometric pressure during the trial was 30.6 inches of mercury, or 15.0 lbs. per square inch. The mean boiler pressure was 80.5 lbs. per square inch by the pressure-gauge in the stoke-hold, and 78.2 lbs. per square inch by the pressure-gauge in the engine-room. The mean vacuum by gauge was 25.4 inches of mercury, corresponding to 2.5 lbs. per square inch of absolute back pressure. The mean pressure in the receiver of the starboard engine was 8.0 lbs. per square inch above the atmosphere, and that of the port engine 8.8 lbs. per square inch above the atmosphere. The mean initial pressures in the high-pressure cylinders were 64.3 lbs. per square inch above the atmosphere in the port engine, and 59.4 lbs. in the starboard engine. The mean back-pressures in the low-pressure cylinders were respectively 4.4 and 4.5 lbs. per square inch absolute.

Boiler Efficiencies.—The mean rate of combustion in the furnaces was 26.1 lbs. of coal per square foot of grate surface per hour, or 0.987 lb. per square foot of total heating surface per hour. The evaporation was at the rate of 7.49 lbs. of water per lb. of coal. The feed entered the boiler at a temperature of 113° Fahr., and the temperature corresponding to the mean boiler pressure was 324° Fahr., so that each lb. of steam must have taken up 1,100 thermal units; hence the equivalent evaporation from and at 212° Fahr. was 8.53 lbs. of water per lb. of coal, or about 9.34 lbs. per lb. of carbon-value in fuel. The equivalent amount of heat utilized per lb. of coal was 8,240 thermal units, or say 62.0 per cent. of the whole calorific value of the fuel, which percentage therefore represents the actual boiler efficiency.

The total calculated calorific value of the fuel burnt per minute was 1,271,000 thermal units.

The weight of dry air per lb. of coal, calculated from the furnace gas analysis, works out to 18.5 lbs., so that the total weight of furnace gases per pound of coal would be about 19.4 lbs. Their specific heat may be taken as 0.238, and they have been raised in temperature from, say, 55° Fahr., the temperature of the outer air, to 835° Fahr., the temperature of the chimney gases. This corresponds to a loss of 3,714 thermal units per lb. of coal, or 28.0 per cent. of the whole calorific value of the fuel, including the raising through the same range of temperature of the steam formed by the combustion of the hydrogen in the fuel. The loss due to the evaporation of the moisture in the fuel works out to 0.4 per cent., and that due to imperfect combustion to 1.3 per cent. These quantities add up to 91.7 per cent. of the whole heat of combustion, leaving a balance of 8.3 per cent., which includes all losses by radiation unaccounted for. The weight of water evaporated from and at 212° Fahr. per square foot of total heating surface was 8.42 lbs. per hour.

The average rate of transmission of heat through the material of the boiler was 8,130 thermal units per square foot of heating surface per hour.

Coal Consumption.—The total coal burnt, namely, 5,742 lbs. per hour, corresponds to 2.90 lbs. per I.H.P. per hour. This is equivalent to 2.65 lbs. of carbon value per I.H.P. per hour.

Engine Efficiencies.—The total I.H.P. being 1,980, and the total feed water used per hour being 43,020 lbs., the feed water used per I.H.P. per hour was 21.73 lbs. The actual heat received by the feed water per minute was 788,700 thermal units, or 398.4 thermal units per I.H.P. per minute, which is 62 per cent. of the whole heat of combustion. The absolute engine efficiency was 10.7 per cent.

Total Efficiency.—The combined efficiency of the boilers and engines, or ratio of the heat turned into work to the total heat of combustion of the fuel, was 0.62×0.107, which is equivalent to 6.6 per cent.

Steam from Indicator Diagrams.—The following are the results of measurements made upon the indicator diagrams taken to ascertain the proportion of steam accounted for by them. The actual weight of feed water used per revolution was 8.28 lbs.:

Proportion of steam accounted for by indicator diagrams.	Lbs. per revolution.	Percentage of total feed.	Percentage present in cylinder as water.
	Lbs.	*Per cent.*	*Per cent.*
Steam present in high-pressure cylinder, port engine, after cut-off, when the pressure was 51 lbs. per square inch above the atmosphere	3.11	71.7	28.3
Steam present in high-pressure cylinder, starboard engine, after cut-off, when the pressure was 51 lbs. per square inch above the atmosphere	2.83		
Steam present in low-pressure cylinder, port engine, near end of expansion, when the pressure was 7 lbs. per square inch below the atmosphere	2.19	53.0	47.0
Steam present in low-pressure cylinder, starboard engine, near end of expansion, when the pressure was 7 lbs. per square inch below the atmosphere	2.20		

Speed of Vessel.—The following notes from the log book of the ship may be of interest:

	Time.	*Distance in nautical miles.*
Left Hull Roads	3.15 A. M.	0
Spurn Light	4.40	24
Dudgeon "	7.45	54
Cromer "	9.5	75½
Cockle "	10.35	97
Corten "	11.15	108
Shipwash "	1.25 P. M.	140
Sunk "	2.20	151
Long Sand Head Light	2.40	158½
Cork Light	4.0	173
Parkeston Quay	4.35	178½

The mean speed between Dudgeon and Cork Lights, which includes nearly the whole of the trial except that portion when

the vessel was slowed down and stopped, was therefore 14.4 knots.

The *Colchester* is a vessel built rather to attain a maximum of power and speed over a short journey than a maximum of economy. Looked at from this point of view, it may be interesting to compare the relation between power developed and the size of the machinery in her case, and in those of the other steamers tested. The net volume of her boilers is about 4,980 cubic feet, or at a rate of 2.5 cubic feet per I.H.P. This ratio in the case of the *Meteor*, *Fusi Yama*, and *Tartar*, is respectively 2.7, 4.53 and 4.34 cubic feet per I.H.P.

3.—TRIAL OF THE S.S. *Tartar*.

Steamer.—This s.s. is owned by Messrs. Gellatly, Hankey, Sewell & Co., and is an excellent example of modern economical engines in a cargo-carrying steamer. She was built in the year 1887 by Messrs. R. Dixon & Co., of Middlesborough, and engined by Messrs. Thomas Richardson & Son, of Hartlepool, with triple-expansion engines, working a single screw. She is a vessel of 332 ft. in length, 38 ft. in breadth, and 27 ft. in depth (moulded). Her gross registered tonnage is 2,389, and she is classed at Lloyd's 100 A1. Her draught during the trial was 8 ft. 6 in. forward, and 15 ft. 6 in. aft, or 12 ft. mean; and her displacement at this draught was 2,250 tons. She was practically without cargo, carrying nothing but water ballast.

The trial was made upon the 27th of November, 1889, on a run from the Thames to Portland. The vessel had arrived in London on the 21st of November, after a voyage from Australia; and the boilers were cleaned, and the engines opened out, and overhauled before the trial, while the ship was unloading in the docks. Steam was got up on the evening of the 26th November, and the vessel went out of dock about 5 A. M., on the 27th. The trial started at 8.25 A. M., as soon as the vessel was far enough down the Thames to run at full speed, and lasted until 6.30 P. M. The weather during the last part of the run was sufficiently rough to cause the engines to race a little,

in consequence of the light draft of the ship. At the time mentioned (6.30 P. M.) the weather had become sufficiently bad to make the taking of accurate observations a matter of considerable difficulty, and it was therefore decided to end the trial, rather than to run any risk of having it spoilt by inaccuracies. In consequence of snowstorms met with going down the Channel, the engines were slowed from 9.52 A. M. to 10.56 A. M., and again for 11 minutes between 12 and 1 o'clock. During these periods the speed was about 64 revolutions per minute, the full speed at other times being about 71 revolutions per minute. Diagrams were taken at the slower speeds, so that they are properly allowed for in the averages. The engines were not stopped at all during the trial, and, with the exception of the intervals named, ran under constant conditions as to position of valve-gear, &c., namely, with the main stop-valve and the throttle-valve fully open, and the link motion slightly linked up.

Engines.—The *Tartar* is fitted with triple-expansion surface-condensing engines, the cylinders being placed in the order, high, low, intermediate, going from forward to aft. The cranks are placed in the sequence—high, low, intermediate. The diameters of the cylinders are 26.03 in., 42.03 in., and 68.95 in., by gauges. The piston rods are all 5.5 in. diameter. There are no tail rods. The stroke of all three cylinders is 3 ft. 6 in. The cylinders are jacketed, and there is a separate steam-admission pipe to each jacket. The clearances of the cylinders are given by the makers as 14.51 and 9.25 and 5.10 per cent. The valve-gear is of the link-motion type, with an independent adjustment for the cut-off. The surface-condenser contains 2,250 square feet of tube surface, in 876 tubes of $\frac{3}{4}$ in. external diameter and of 13 ft. .0$\frac{7}{8}$ in. length between tube-plates. The screw propeller has four blades, and is 16 ft. 6 in. diameter, with a pitch of 18 ft.

Boilers.—Steam is supplied by two double-ended steel boilers with Fox's flues, designed to work at a pressure of 150 lbs. per square inch. The boilers are 13 ft. in mean diameter, and 14 ft. 9 in. long, each with two furnaces at each end, there being eight furnaces in all. The total grate surface is 161 square feet, each grate being 5 ft. 6 in. in length, and 3 ft. 8 in. broad. The total

heating surface is 5,226 square feet, of which 4,366 square feet is tube surface. The total heating surface is, therefore, 32.5 times, and the tube surface 27.1 times, the grate surface. There are 992 tubes, 3 in. diameter outside, and 5 ft. 9 in. long over plates. The internal diameter of the funnel is 7 ft., and its total height 57 ft., above the center of the furnaces.

The total weight of the engines and boilers, including water in condenser, pipes and boilers, and all mountings, is about 191 tons. The net volume of the boilers is about 4,710 cubic feet.

Duration of Trial.—The duration of the trial from start to finish was 10 hours 5 minutes, or 605 minutes.

Coal Measurement.—The coal was weighed as in the other cases, from 500 to 600 lbs. at a time being put down on each side of each stoke-hold. The trial was started with clean floors, and the time of first stoking from each weighed lot of coal was noted, as well as the time when the whole of the weighed coal was finished. The fires were not cleaned during the run. The ashes were weighed after the trial was over, the ash-pits having been cleaned a little time after it commenced. The coal used was Welsh Penrikyber (Glamorganshire) throughout. Coal samples were taken frequently during the run from both stoke-holds, and the final analysis, after a thorough mixture of all these samples, is as follows:

	Coal as used.	*Dry Coal.*
Carbon	87.98 per cent.	88.93 per cent.
Hydrogen	4.22 "	4.26 "
Moisture	1.07 "	0.00 "
Ash	3.42 "	3.46 "
Nitrogen, Sulphur, Oxygen, &c., by difference	3.31 "	3.35 "
	100.00	100.00

A sample of the ash was also collected, and has been analysed by Mr. Wilson, as follows:

Loss on ignition (= Carbon)	65.53 per cent.
Mineral matter	34.47 "
	100.00

The calculated calorific value of the fuel is 14,995 thermal units, which corresponds to the evaporation of 15.52 lbs. of water from and at 212° Fahr., and to an equivalent carbon value of 1.031 lbs. per lb. The total coal used was as follows:

After stoke-hold,	port side	5,322	lbs.
" "	starboard side	5,052	"
Forward "	port side	4,701	"
" "	starboard side	4,162	"
		19,237	"

This amounts to 31.8 lbs. per minute, or 1,908 lbs. per hour. As a check upon this consumption, and in order to see how far it might be affected by any irregularities in measurement at the end of the trial, the consumption has also been worked out over a time (in average 560 minutes) at the beginning and end of which the condition of the stoke-hold floors, and of the fires, was identical, and for this interval the coal consumption works out to 32.08 lbs. per minute. It has been thought wise to assume, therefore, that the actual coal consumption was 32.0 lbs. per minute, or 1,920 lbs. per hour; the small difference between this and the foregoing figures being probably caused by the increasing difficulty of measurement at the very end of the trial. The total amount of ash taken from the two stoke-holds at the end of the trial was 291 lbs., but the ash-pits were not cleaned out until some little time after the trial commenced, so that this weight corresponds only to a period of 6 hours 53 minutes. The ash was, therefore, equivalent to 0.70 lbs. per minute, or 42 lbs. per hour; which is 2.2 per cent. of the total weight of the fuel put on the bars.

Furnace Gases.—A very complete series of furnace gas samples was collected, a sample being collected over mercury by Mr. Wilson every half-hour. The analyses of these samples are given in Table 4.

The chimney temperatures were read every quarter of an hour throughout the trial from a mercury thermometer as before. The average reading was 477° Fahr. It remained very fairly constant during the whole trial.

The chimney draft was read on a U gauge at the same time that the temperature was taken, and varied from $\frac{3}{16}$ in. to $\frac{1}{4}$ in. of water.

Feed-Water Measurement.—The feed water was measured in the same tanks used for the *Colchester*. Each tank lasted in this case from 5 to 6 minutes. The total amount of water used was 217,430 lbs., over a total time of 605 minutes. This amounts to 359.4 lbs. per minute, or 21,564 lbs. per hour, but reasons are given later on for supposing that a considerable percentage of this quantity was not actually turned into steam. At first pumping was carried out by one of the feed-pumps of the engine, but for the last four hours a donkey pump was used, as it was found to give much less trouble, and to be much more

TABLE 4.

"TARTAR" TRIAL.

Analyses of Funnel Gases by Volume.

No. of Sample.	Carbonic Acid.	Carbonic Oxide.	Oxygen.	Nitrogen.	Time of Collecting.
	Per cent.	Per cent.	Per cent.	Per cent.	
1	7.28		12.45	80.27	8.30 A.M.
2	7.61		11.88	80.51	9.0 "
3	8.22		11.38	80.40	9.30 "
4	6.97		13.20	79 83	10.0 "
5	5.91		14.32	79.77	10.30 "
6	6.24		13.38	80.38	11.0 "
7	6.94		13.10	79.96	11.30 "
8	6.44		13.80	79.76	12 noon.
9	7.70		12.10	80.20	12.30 P.M.
10	6.21		13.57	80.22	1.0 "
11	8.24	0.00	12.00	79.76	1.30 "
12	5.62		14.40	79.98	2.0 "
13	6.29		13.15	80.56	2.30 "
14	6.79		12.78	80.43	3.0 "
15	6.80		13.30	79.90	3.30 "
16	6.54		12.71	80.75	4.0 "
17	6.95		13.10	79.95	4.30 "
18	6.03		13.08	80.89	5.0 "
19	6.67		13.17	80.16	5.30 "
20	6.61		13.52	79.87	6.0 "
21	5.00		15.27	79.73	6.30 "
Mean	6.72	0.00	13.12	80.16	

"TARTAR" TRIAL.

Analyses of Funnel Gases by Weight.

No. of Sample.	Carbonic Acid.	Carbonic Oxide.	Oxygen.	Nitrogen.	Time of Collecting.
	Per cent.	Per cent.	Per cent.	Per cent.	
1	10.80		13.43	75.77	8.30 A. M.
2	11.28		12.80	75.92	9.0 "
3	12.15		12.23	75.62	9.30 "
4	10.34		14.25	75.41	10.0 "
5	8.81		15.52	75.67	10.30 "
6	9.30		14.50	76.20	11.0 "
7	10.30		14.15	75.55	11.30 "
8	9.58		14.93	75.49	12 noon.
9	11.40		13.03	75.57	12.30 P. M.
10	9.25		14.70	76.05	1.0 "
11	12.17	0.00	12.89	74.94	1.30 "
12	8.39		15.63	75.98	2.0 "
13	9.37		14.25	76.38	2.30 "
14	10.09		13.82	76.09	3.0 "
15	10.10		14.37	75.53	3.30 "
16	9.74		13.76	76.50	4.0 "
17	10.32		14.14	75.54	4.30 "
18	9.00		14.19	76.81	5.0 "
19	9.92		14.24	75.84	5.30 "
20	9.83		14.61	75.56	6.0 "
21	7.48		16 61	75.91	6.30 "
Mean	9.98	0.00	14.19	75.83	

conveniently under control. The steam for this donkey-pump was supplied from the main boilers. The steam made by the boilers all went to the main engines, steam being kept up in the donkey boiler for all auxiliary engines; and the pipes and connections having been carefully examined, in order to see that no unintentional communications existed. The water level in the boilers was, on the average, 1½ in. lower at the finish than at the start of the trial. This corresponds to a total of 1,874 lbs. of water, or about 0.86 per cent. of the whole feed; this amount has been added to the quantities given above. It may be interesting to notice that an error of half an inch in reading the level in one boiler in a case like this would make a difference of about 375 lbs., or under 0.2 per cent. of the water used.

Power Measurement.—Indicator diagrams were taken every half hour from each end of each cylinder as before. There were

thus 21 complete sets of diagrams, or 126 single cards taken in all. The indicators used were those lent by Messrs. McInnes & Cairns, which were used also for the *Colchester*. The following are the mean effective pressures in the cylinders in lbs. per square inch :—

Cylinder.	*Top.*	*Bottom.*	*Mean.*
High-pressure	35.74	38.04	36.89
Intermediate	19.92	20.23	20.07
Low-pressure	7.19	7.18	7.18

These pressures correspond to the following I.H.P.'s:—

High-pressure cylinder	283.7
Intermediate cylinder	408.5
Low-pressure cylinder	395.2
Total indicated horse-power	1,087.4

The maximum I.H.P. given by any one set of cards was 1,296.7, which occurred at 8:45 A. M., with 73.2 revolutions per minute, and 146 lbs. of steam. The minimum I.H.P. by any one set of cards was 719.3, at 10:10 A. M., with 61.4 revolutions per minute, and 135 lbs. of steam. From the 126 diagrams which were taken during the trial, a mean diagram has been plotted for each of the three cylinders.

Speed.—The counter was read every quarter of an hour. The total number of revolutions made was 42,350 in 605 minutes, or exactly 70 revolutions per minute. The maximum number of revolutions per minute for any quarter of an hour at full speed was 73.8, and the minimum 70.3 revolutions per minute; while the average speed during the time that the engine was slowed down was 63.7 revolutions per minute.

Steam Jackets.—As has been already mentioned, each jacket received steam through a separate valve. The connections from the jacket-drains to the hot-well were broken, and all three jackets were made to drain into a small tank placed at the back of the engine; and the rate at which the jacket-water collected in this tank was measured 14 times during the trial, on most occasions for nearly a quarter of an hour continuously. The water collected in this way amounted in all to 2,476 lbs. in 175

minutes, or 14.15 lbs. per minute, which is 3.94 per cent. of the total feed. This quantity is, of course, included in the total feed measured through the tanks. It was not possible to jacket the high-pressure cylinder with steam of full boiler pressure; and under these circumstances it did not appear advisable to allow any steam to pass into that jacket, which was accordingly shut off. The quantity of steam given is therefore from the intermediate and low-pressure cylinder jackets only.

Pressures, &c.—The mean barometric pressure during the trial was 29.6 inches of mercury, or 14.6 lbs. per square inch. The mean boiler pressure was 143.6 lbs. per square inch by the pressure gauge in the stoke-hold, and 140.4 by that at the engine. The pressures in the high-pressure, intermediate, and low-pressure valve-chests, were respectively 131.4, 46.1, and 3.8 lbs. per square inch above the atmosphere respectively. The pressures in the steam-jackets of the intermediate and low-pressure cylinders were 52.5 and 14.0 lbs. per square inch respectively. The cock admitting steam into the high-pressure jacket, although nominally shut, allowed enough steam to leak through to enable the gauge to show pressures varying from 0 to over 50 lbs. per square inch. The actual amount of water coming from the high-pressure jacket-drain, however, was too small to affect appreciably the quantities given in the last paragraph. The mean vacuum was 26.2 in. of mercury by gauge, which corresponds to a mean absolute back-pressure in the condenser of 1.7 lbs. per square inch. The mean initial pressure in the high-pressure cylinder was 121.8 lbs. per square inch above the atmosphere; and the mean back-pressure in the low-pressure cylinder was 10.5 lbs. per square inch below the atmosphere, which corresponds to an absolute back pressure of 4.1 lbs. per square inch in the cylinder.

Boiler Efficiencies.—The mean rate of combustion in the furnaces was 11.93 lbs. of coal per square foot of grate surface per hour, or 0.367 lbs. per square foot of total heating surface per hour. The total amount of feed-water pumped into the boilers was at the rate of 11.23 lbs. per lb. of coal. As the feed entered the boiler at an average temperature of 101° Fahr., and the tem-

perature corresponding to the mean boiler pressure was 362° Fahr., each lb. of steam must have taken up 1,123 thermal units; and the equivalent evaporation from and at 212° Fahr., had the whole of the feed-water been turned into steam, would therefore have reached the very high figure of 13.06 lbs. of water per lb. of coal. The equivalent amount of heat utilized per lb. of coal would be 12,610 thermal units, or say 84.3 per cent. of the whole calorific value of the fuel. Such a figure, although unusually high, is of course not in the nature of things impossible; but if it really represented the evaporation in this case, it would follow (as will be seen from the figures below) that the unusual economy of the boilers was accompanied by an unusual want of economy in the engines. Fortunately the measurements made were sufficiently complete to enable it to be stated positively that (from whatever cause) the whole of the water pumped into the boiler was *not* turned into steam. The observations on which this statement is based are summarised in the next paragraph.

The very complete and uniform series of samples of chimney gases, of which the analyses have been given in Table 4, enable the calculations connected with them to be made with considerable exactness and certainty. The weight of dry air per lb. of coal calculated from these analyses works out to 31.6 lbs., so that the total weight of furnace gases per lb. of coal would be about 32.6 lbs. The gases have been raised in temperature from the temperature of the outer air, say about 55° Fahr., to 477° Fahr., the chimney temperature. Taking the mean specific heat of the gases as 0.238, this corresponds to a loss of 3,307 thermal units per lb. of coal, or 22.1 per cent. of the whole calorific value of the fuel, including the raising through the same range of temperature of the steam formed by the combustion of the hydrogen in the fuel. The loss due to evaporation of the moisture in the fuel is so small as to be negligible, and in no one of the twenty analyses was any carbonic oxide found to be present in the chimney gases. The loss by radiation was no doubt considerably less than in the *Colchester* (the stoke-holds being very much cooler), but taking it as low as 8 per cent. and adding to it the 22 per cent. carried away by the chimney gases, it seems

certain that the maximum boiler efficiency which can have been reached cannot have exceeded 70 per cent. This would correspond to an evaporation of 10.8 lbs. of water from and at 212° Fahr. per lb. of coal, which is about 83 per cent. of the apparent evaporation mentioned above. It is moreover to be noted that on the original log sheets the reporter finds in five different places memoranda made at different times by different assistants who were taking indicator diagrams, that they were unusually troubled by the amount of water which came through the indicator cocks.

Coal Consumption.—The total coal burnt, namely 1,920 lbs. per hour, corresponds to 1.77 lbs. per I.H.P. per hour. This is equivalent to 1.82 lbs. of carbon-value per I.H.P. per hour.

Engine Efficiencies.—Although the total quantity of water pumped into the boilers per I.H.P. per hour amounted to $\frac{21,564}{1,087.4}$ or at the rate of 19.83 lbs. per I.H.P. per hour, the quantity of *steam* consumed could not have exceeded, as the figures in the preceding paragraph show, 16.5 lbs. per I.H.P. per hour. The difference must be attributed to priming.

Total Efficiency.—The combined efficiency of the boiler and engines, or ratio of the heat turned into work to the total heat of combustion, was 9.7 per cent.

Steam from Indicator Diagrams.—The following are the results of measurements made upon all the indicator diagrams taken, to ascertain the proportion of steam accounted for by them. The actual weight of feed-water used per revolution was 51.3 lbs.

Proportion of steam accounted for by indicator diagrams.	Lbs. per revolution.	Percentage of total feed.	Percentage in jackets or present in cylinder as water.
Steam present in high-pressure cylinder after cut-off, when the pressure was 115.4 lbs. per square inch above the atmosphere	*Lbs.* 2.32	*Per cent.* 45.2	*Per cent.* 54.8
Steam present in intermediate cylinder when the pressure was 33.4 lbs. per square inch above the atmosphere	2.53	49.2	50.8
Steam present in intermediate cylinder when the pressure was 9.4 lbs. per square inch above the atmosphere	3.03	59.0	41.0
Steam present in low pressure cylinder near end of expansion, when the pressure was 7.1 lbs. per square inch below the atmosphere	3.12	60.7	39.3

Speed of Vessel.—The following notes from the log-book of the ship may be of interest:—

	Time.
Left London Docks	5:0 A. M.
Gravesend	7:20
Nore	8:35
Girdler Lightship	9:30
Dover	12:40 P. M.
Dungeness	2:20
Royal S. Light-vessel	4:20
Beachy Head	5:0
Owers	8:0
St. Catharine	11:30
Portland	5:0 A. M.

The Research Committee on Marine Engine Trials, in presenting to the members of the Institution their second report upon their work, desire it to be quite understood that they do not at all look upon their duties as yet ended. They are glad to be able to think, however, that they have already done enough to show that engine trials can be conducted on board ship on as complete a scale and in as scientific a method as on land (although perhaps with greater difficulty and discomfort to the observers), and that this can be done without any undue interference with the normal working of the ship or engines.

The committee wish also to say that they hope, when their work is completed, to present to the members a comparative summary of all their experiments, in the discussion on which many points as to the influence of design on economy may be better discussed than at present. Their future work may be much helped, however, by suggestions made at this stage as to improvements in the methods or in the instruments used, or as to the best means of ensuring the accuracy, or measuring the errors of gauges, indicators, &c. They would also point out that there still does not exist any satisfactory means for measuring the *effective* H.P. of a marine engine in its actual work, *i. e.*, the power actually expended in pushing the ship forward (as measured by the thrust of a propeller and the speed of the ship), nor for measuring the quantity of the condensing water, a knowledge of which is necessary to enable a "heat balance" for the engines to be made.

There is added to this report a table showing the comparative results obtained in the four trials as yet made by the committee, and also an appendix giving the instructions for taking indicator diagrams now adopted.

Appendix I.

Instructions given for taking Indicator Diagrams.

See that indicators have proper length of stroke, and are not on stops.

Blow through before taking every card.

Mark number of card in top right-hand corner.

Let H.P. top cards begin with No. 1, H.P. bottom with 101, I.P. top with 201, I.P. bottom with 301, L.P. Top with 401, and L.P. bottom with 501.

Let top and bottom cards of each cylinder be taken as nearly as possible together, and the whole set as quickly as possible.

After having taken a set of cards and entered time in log, &c., put on the new papers at once for the next set.

Draw atmospheric line by hand with full stroke.

COMPARATIVE RESULTS OF THE TRIALS OF FOUR STEAMERS, *METEOR*, *FUSI YAMA*, *COLCHESTER*, AND *TARTAR*.

		Meteor.	*Fusi Yama.*	*Colchester.*	*Tartar.*
1	Name of Vessel				
2	Date of trial	24th June, 1888.	14 & 15 Nov., 1888.	9 Nov., 1889.	27 Nov., 1889.
3	Duration of trial, hours	17.15	13.95	10.88	10.08
4	Type of engines	triple	compound	twin compound	triple
5	Cylinder diameter, high pressure, inches	29.37	27.35	(two) 30	26.03
6	" " intermediate, inches	44.03	—	—	42.03
7	" " low pressure, inches	70.12	50.3	(two) 57	68.95
8	Stroke, length, inches	47.94	33	36	42
9	Boilers, number of main boilers	2	1	2	2
10	" single-ended or double-ended	double	single	double	double
11	Furnaces, total number	12	3	12	8
12	Heating surface, total, square feet	6,648	2,257	5,820	5,226
13	" " tubes, square feet	5,760	1,689	4,770	4,866
14	Grate area, square feet	208	52	220	161
15	Total heating surface to grate area, ratio	32.0	43 4	26.5	32.5
16	Tube surface to grate area, ratio	27.7	32.5	21.7	27.1
17	Grate area to flue area through tubes, ratio	—	4.05	5.51	4.50
18	" area through funnel, ratio	5.04	3.21	4.77	4.19
19	Mean boiler-pressure above atmosphere, lbs. per sq. inch	145.2	56.84	80.5	143.6
20	" admission pressure high-pr. cyl. above atmos, lbs. per sq. inch	134.4	50.3	64.3 59.4	121.4
21	" effective " high-pr. cyl, lbs. per sq. inch	58.46	30.74	45.65 42 07	36.89
22	" " " inter. " lbs. per sq. inch	19.50	—	—	20.07
23	" " " low pr. " lbs. per sq. inch	12.38	10.87	13 42 12.42	7.18
24	" " " total reduced to low-pr. cyl, lbs. per sq. inch	29.9	19.9	24.8	19.8
25	" exhaust " low pr. cyl. below atmos, lbs. per sq. inch	11.6	10.9	10.6 10.5	10.5
26	" vacuum in condenser below atmos, lbs. per sq. inch	12.17	12.48	12.49	12.9
27	Revolutions per minute, mean, revs.	71.78	55.59	86.0 87.1	70.0
28	Indicated horse-power, mean total, i.h.p.	1,994	371.3	1,022.5 957.2	1,087.4

28	Indicated horse-power, mean total	1,994	371.3	{ 957.2 }	1,007.4
29	Coal burnt per minute, lbs.	66.75	16.45	95.7	32.0
30	" per hour, lbs.	4,005	987	5,742	1,920
31	" per sq. ft. of grate per hour, lbs.	19.25	18.98	26.1	11.93
32	" per sq. ft. of total heating surface per hour, lbs.	0.602	0.487	0.987	0.367
33	" per indicated horse-power per hour, lbs.	2.01	2.66	2.90	1.77
34	Carbon-value of 1 lb. of coal as used, lbs.	0.878	0.878	0.913	1.031
35	" equivalent per i.h.p. per hour, lbs.	1.76	2.33	2.65	1.82
36	Feed-water per minute, lbs.	497.7	131	717	359.4
37	" per hour, lbs.	29,860	7,860	43,020	21,564
38	" per sq. ft. of total heating surface per hour, lbs.	4.49	3.48	7.39	4.13
39	" per lb. of coal, lbs.	7.46	7.96	7.49	[11.23]
40	" per lb. of coal from and at 212° F., lbs.	8 21	8.87	8.53	[13.06]
41	" per lb. of carbon value from and at 212° F., lbs.	9.62	10.10	9.34	[12.67]
42	" per indicated horse-power per hour, lbs.	14.98	21.17	21.73	[19.83]
43	Calorific-value of 1 lb. of coal as used, Th. U.	12,770	12,760	13,280	14,995
44	Percentage of line 43 taken up by feed-water	62.0	67.2	62.0	—
45	" " carried away by furnace gases	21.9	23.5	28.0	22.1
46	" " lost by imperfect combustion	3.6	0.0	1.3	0.0
47	" " expended in evaporating moisture in coal	1.2	0.9	0.4	0.0
48	" " unaccounted for	11.3	8.4	8.3	—
49	Heat taken up by feed-water per minute, Th. U.	528,600	141,100	788,700	[403,600]
50	" turned into work per minute, Th. U.	85,240	15,870	84,630	46,490
51	" taken up by feed-water per i.h.p. per minute, Th. U.	265.6	380	398.4	[371.2]
52	Efficiency of boiler (line 44), per cent.	62.0	67.2	62.0	—
53	" of engine (line 50 ÷ line 49), per cent.	16.1	11.2	10.7	[11.5]
54	" of engine and boiler combined (line 52 × line 53), per cent.	10.0	7.6	6.6	9.7
55	Mean velocity of steam through water-surface in boilers per minute, feet	—	6 28	8.6	3.43
56	Space occupied by boilers per i.h.p., cubic feet	2.72	4.53	2.52	4.33
57	Weight of engines, boilers, &c., with water, per i.h.p., tons	0.20	0.27	0.20	0.27
58	Clearance surface, high-pressure cylinder, square feet	—	—	{ 22.6 22.6 }	—
59	" " intermediate, square feet	—	—	—	—
60	" " low-pressure, square feet	—	—	{ 56.9 56.9 }	—
61	Speed of vessel, mean, during trial, knots	14.6	—	14.4	—

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First watch to get indicators ready before starting, and note numbers of indicators and marks and scales of springs on log sheet.

Second watch to verify these numbers and after taking last cards on the watch interchange top and bottom indicators of each cylinder.

Third watch to verify the changed numbers.

Fourth watch, after taking last set of cards, to take off indicators and put them in boxes.

Examine every set of cards before putting away, to detect anomalies and errors. Compare top and bottom cards of each cylinder for length, pressure and general appearance. If any difference is noticed, or anything suspicious about cards, report at once to engineer in charge of watch.

Keep the indicator boxes shut, and the papers in them.

TUBULOUS BOILERS.

[Continuation of the discussion of the paper on this subject by Assistant Engineer S. H. LEONARD, U. S. Navy, p. 163.]

MR. MIERS CORYELL, ASSOCIATE. — *The Belleville Boiler.* — Assistant Engineer S. H. Leonard's paper on Tubulous Boilers in the last number of the JOURNAL is very interesting; my endeavor will be to supplement his contribution by some additional detailed facts relative to the Belleville boiler. He appreciates what should be obtained from tubulous boilers by giving a comprehensive statement of the requisites: 1st, a higher pressure; 2d, less weight; 3d, rapidity of generating steam; 4th, facility in making repairs; 5th, a positive and regular alimentation; 6th, pure water or its purification before entering the tubes; 7th, care or the facility of cleansing the heating surface on both the water and fire sides; 8th, durability and consequent immunity from accident during the life of the boiler.

It is unfortunate, I think, that the Belleville type of boiler is not better known to the Naval service by personal acquaintance and management. Its employment in steamships working with large horse power at sea and on long voyages for the past ten years has attracted some attention, but owing to the limited amount of data attainable, the interest taken in it has been academic rather than practical, on account of an erroneous impression of its real capabilities based on the test made in November, 1887, on the boiler of the yacht *Shearwater*.

Of this test I wish to say something. The designed values for grate area, heating surface, ratio and weight were 31.43 square feet; 824 square feet; 1 to 26.27; and 29,700 pounds, respectively, instead of those given, which, of course, were correct for the trial conditions. The *Shearwater's* boiler is arranged for a discharge pressure of 150 pounds, but only 111 pounds were used at test, at which the small degree of moisture shown is possible, whereas a considerable temperature above that due to this pressure (111 pounds) would have been recorded had the boiler pressure been maintained at 150 pounds, as was intended. The combustion per square foot of grate was too small to test the efficiency of any type of boiler.

The incompleteness of the boiler test by the experimentors may be inferred from the fact that there were presented to their observation in the construction of this boiler three very important conditions, viz: 1st, positive circulation of the water, varying in speed of flow according to the intensity of the heat of the furnace; 2d, the working of the automatic feed regulator; 3d, the precipitation of the calcareous matter contained in the feed water into a receiver, which is thus prevented from being taken up by the water of alimentation. That is to say, in using salt water alimentation, no scale is made in the tubes with a saturation as high as the 6th or 7th degree (full saturation at the 10th degree); and again, by adding lime to the water from the fresh water condenser, the residuum from the lubricants will also be carried into the sediment chamber and not passed through the tubes.

Considering the importance which has been given to this test and the influence it has had in establishing wrong conclusions,

I may be pardoned for characterizing it as not a typical case of the Belleville boiler system, and, in fact, an inadequate presentation of it, as I hope to be able to prove in this paper. (The report of the test I am referring to is given in the report of the Engineer-in-Chief of the Navy for 1888, pp. 39–48.)

The Belleville boiler answers without an argument the first requisite, *high pressure.* Being made for a twenty years' life, it is of a durable thickness at its heating surfaces and attachments, and capable of a pressure up to 600 pounds and over.

2d requisite, *less weight.*—In the table of comparative conditions this boiler is compared with tubulous boilers of such thin material as to sacrifice durability, and directly their safety. It is also in comparison with a boiler of the Scotch type, to carry only 70 pounds. Compared with the tubulous boilers, a Belleville boiler may be constructed of as light material and compare favorably in weight, its durability being thereby lessened to their standard. It is not, however, possible to increase the material of their tubes to a thickness to resist the effects of pitting, which is a prime requisite, considering that their construction does not free the feed water from acids which cause the pitting. Compared with the Scotch boiler, construct that boiler of a thickness of material to withstand only 300 pounds steam pressure, as against the 600 pounds of the Belleville, to make the comparison, then the lighter weight will be largely in favor of the Belleville system.

3d requisite, *rapidity of generating steam.*—Firing with wood or like inflammable material, it requires only 18 minutes from cold water to 150 pounds pressure; with banked fires, it is 7 minutes to 150 pounds; firing with coal, from cold water the ordinary time is from 22 to 30 minutes, according to the substance used for the ignition of the coal and the effort made. Being a coil boiler, the flow of the circulating current is continuous in one direction of circuit, and is as the heat applied to the tubes. It follows that the hotter the fire the more water is presented to the heat to carry it off. This has been the subject of much experimental study by Mr. Belleville, resulting in the information that up to 44 pounds combustion per square foot of

grate per hour, the ratio of 26 to 1 may be relied upon, in his generator, to yield about the same comparative vaporizing power per pound of coal. This proves the circulation to be as the heat, and demonstrates also the fact that the conditions of the Belleville furnace are excellent, serving to thoroughly mix the gases by the mingling of the heat and gas-laden currents among the tubes, "successive ripplings," as Mr. Belleville terms it.

4th, *facility in making repairs.*—This is evident without argument.

5th, *positive alimentation.*—The Belleville automatic regulator of the feed water supplies alimentation, in connection with a special pump, *according to the need*, which is very important in all tubulous types. My own experience is that it is a reliable instrument. No personal attention can approach it for efficiency because of its sensitiveness in regard to the water level. It is not intended to do away with the attendant's supervision, but it assists him greatly. I have watched its operation at sea and observed its perfect control over the pump discharge, and am certain no engineer once using it will ever wish to go on a voyage without its being one of his boiler attachments. It never ceases its attention, and from all the information respecting its working, from engineers who have had years of experience with it, I have the opinion that it is a reliable machine and has never failed to work properly.

6th, *pure water.*—The discovery of the Belgian chemist (Cousté) that water containing calcareous matter, if suddenly raised to a sufficiently high temperature, causes that matter to become insoluble, and by its superior gravity to be precipitated, was made practical use of in Mr. Belleville's boilers of 1877, and since then their application to marine purposes has been rapid in both the naval and mercantile marine. His studies in this direction have resulted in the application of an attachment, without complexity of design, which, using the circulating flow as a medium, with the greater specific gravity of the precipitated matter as auxiliary, causes a regular deposition of the calcareous matter in a sediment chamber, where it cannot be again taken up by the water of alimentation.

7th, *care.*—It is apparent from the engraving of the Belleville boiler, and Assistant Engineer Leonard's description, that there is no form of boiler so well equipped with facilities looking to its care. This is further explained by saying that there is no part of the generator or its attachments inaccessible to the work of cleansing, either inside or outside, and this can be done quickly without loss of time in preparation, and, quite as important, the completeness of this inside or outside cleansing can be ascertained by a sight inspection from the fire room with the aid of a lighted candle at the end of a forked iron rod.

The openings are covered with caps at every point that needs inspection or attention, which also afford facility for possible repairs to the tube elements that constitute the fire surface, which may be withdrawn into fire rooms by unscrewing three nuts. As all parts are manufactured on the interchangeable system, the element may be replaced by another kept in spare, or be put back itself. Experimentally, a boiler has been shut off a group, the water discharged after withdrawing the fire, an element taken out and put back in place, the boiler filled again, and 150 pounds steam pressure had in 25 minutes.

8th, *durability.*—This results from the care which can be given to the boiler inside and outside, as above stated; and, owing to the careful design and construction of the Belleville boiler, there is a perfect drainage, so that forcing water through it by the hose sweeps out all obstructions. Still, as it is possible by carelessness to deposit acids which will cause pitting, there must be provided sufficient material to resist its effects in all boilers claiming durability.

It must be conceded that the Belleville generator has been used for marine purposes long enough to obtain accurate information concerning it. I have an experience extending through seven years, and I am in correspondence with intelligent engineers who have charge of groups developing large horse power on merchant steamers making long voyages. The latest is *L'Australien*, which has just completed a round voyage from Marseilles to Sydney. The piston pressure used is 210 pounds; the boiler pressure about 260; I. H. P., 7,400. The

engineer writes me: "Have had excellent results from engines and boilers; the boilers have given no trouble whatever; the ship has developed great speed; the economy exceeds all expectations."

From the above quotation, the Belleville boilers have achieved another signal success, which places this tubulous type far ahead of all other types, when the high pressure used, the large power developed, and the long sea voyage are considered.

Mr. Belleville, in his boiler designs, carefully proportions the opening at the base of the tube elements where the feed water enters according to the power of the vaporization required; it is important to do so to secure a proper distribution of the water among the tubes. In all other boilers, the locality of the greatest heat in a furnace changes as the conditions favoring a better combustion shift from one part to another; in the tubulous type the water flow is increased in the tubes nearest to this section of the furnace, to be withdrawn from the others, resulting in their damage before the water level can be restored.

The Belleville furnace, however, provides for the complete admixture of the gases within it, by the action of the jets of steam named "mixer of gases," which preserve a uniform heating in the furnace, and the placing of the obstructive plates so as to cause the "ripplings" of the heat currents among the tubes.

The application of large boiler power in a vessel on the Belleville system is effected by grouping the generators, preferably of about 30 square feet of grate each, placing them back to back with their center line over the keel line, having a fire room at each wing, thus giving each group two fire rooms, with the coal bunkers outboard of them. This provides excellent ventilation, and locates the bunkers directly behind the firemen, so as to require but little trimming of the coal; also opposite each furnace door there is an opening into the bunkers, making coal handling very easy and rapid.

The fire rooms may be fitted closed and air tight, but it is a better plan to force the air into closed ash-pans with damper regulators, which permits of using natural draft in some furnaces in connection with forced draft in others, while the mod-

erate use of a jet in the smoke-pipe supplements usefully either of the other means of getting draft.

There are in the management of the Belleville boilers at sea some excellent features of practice, which tend to preserve the steaming efficiency unimpaired during the whole of a long voyage. One is that the generators are never allowed to become clogged with soot or ashes. The cleaning is so readily and speedily done as to give no excuse for the reverse; neither need the outside or fire surface ever become coated with a non-conducting substance from the use of bituminous coal, because a special tool, used from the fire room, clears it from the tubes in a few minutes for each generator.

There is a special cleaning at short periods while on the voyage of each entire generator in turn inside and outside, resulting in the boilers coming into port at the end of a voyage ready to begin another at once. There are other things new to engineers regarding boiler management where the Belleville generators are used, well worth investigation. The manipulation is by a routine of work, each operation leading up to the next without a special order, which, while not fatiguing, results in what may be termed a discipline in the work of the fire room, producing a regular firing and steady steam production, which is most desirable in order to attain the higest duty with economy, as well as to develop the greatest efficiency.

To determine the best size of a furnace to yield the greatest efficiency when forced draft is used, it has been experimentally decided by Mr. Belleville, that grates five feet long and six feet wide in a furnace having two doors, give the best results, for the reason that one-half can be worked at a time, to advantage in the regular heating; that is, one-half is in full power of heat while the other may be progressing toward it. This is where the regularity or discipline of the fireman's work is most to be appreciated, as the furnaces are fired successively, one-half at a time, at longer or shorter intervals according to the want.

In the Bellville generators, the vaporization is reduced or entirely stopped by opening the tube box doors and keeping closed the furnace doors and ash pit dampers, thus preventing the

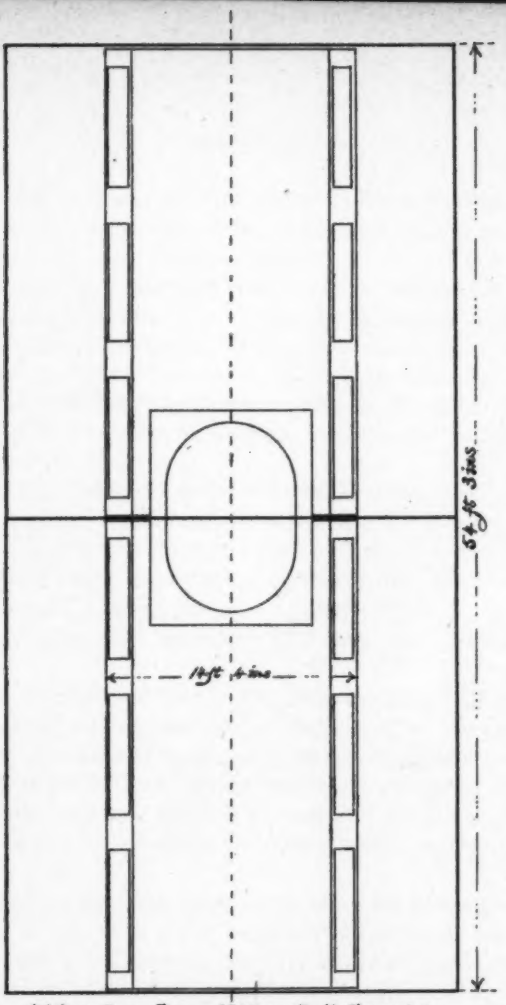
strains upon the material of the generator due to the introduction of cold air below the tubes; this precaution is not, so far as I know, taken in other tubulous boilers.

In all tubulous boilers, the best arrangement of the tubes is with a little inclination upward from the horizontal, by which the water is always presented to the greatest heat, which is along the bottom of the tube. The return bends, as made in the Belleville boiler, have the effect to partially separate the steam from the water and thus assist in keeping the water at the bottom of the tube.

The vertical water tube under severe heat exposure, will have the water in the center and the rising steam bubbles at its circumference, and cannot therefore carry off the heat with the same comparative efficiency with the higher degrees of furnace heat due to forced combustion; consequently the ratio of heating surface to grate area must be larger than with the horizontal tubes.

It is to be observed that in the Belleville boiler the water entering the tube elements becomes before and at the exit at the top in the condition of foam, and is projected against the specially arranged division plates within the separator (steam drum), to the end that saturated steam is presented at the discharge outlet to be reduced in pressure and super-heated at the reducing valve.

This condition of the water within the tube elements renders the use of the feed water regulator a matter of great convenience and safety, and the introduction of this instrument became a matter of importance to Mr. Belleville, because with this device, as constructed by him, the real water level is the same as the apparent level shown in the glass. That is to say, withdraw the fires and stop the vaporization, and the real level coincides with the apparent level, so that the float which controls the admission of the alimentary water is the true index of the real water level, while the tubes, as explained, when subjected to heat are filled with water and foam, the real level is correctly registered by the lever of the regulator attached to the float.

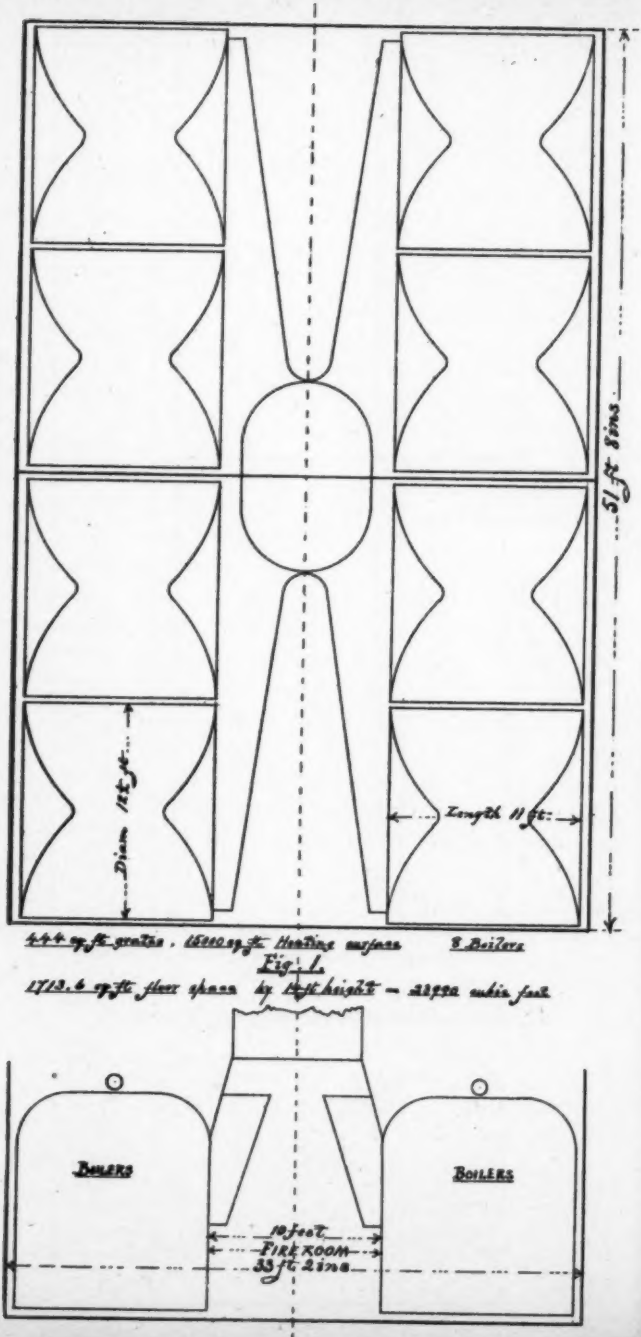


444 sq. ft. grates. 15,000 sq. ft. Heating surface.

Fig. 2.

12 Generators

1401.45 sq. ft. floor space by 18 ft. height = 19420 cubic feet.


51 ft 8 ins
Diam 12 ft
Length 11 ft.
444 sq ft grates, 18000 sq ft Heating surface 8 Boilers
Fig. 1.
1713.6 sq ft floor space by 28 ft height = 48000 cubic feet
BOILERS
BOILERS
10 feet
FIRE ROOM
33 ft 2 ins

The rise and fall of the water in the glass tube is not important so long as the automatic regulator indicates safety. The rise and fall of the water in the glass, if over an inch of movement, shows that the fire is not well served to give a regular heating, and has no other significance, but serves to warn the attendant of a want of attention to the fires.

To illustrate the comparative weight and space between a battery of Scotch type boilers and a group of Belleville generators for the same power, I present two diagrams, figures 1 and 2. Figure 1 is copied from the report of Bureau of Steam Engineering for 1889, plates 22 and 25, designed for proposed *Armored Cruising Monitor*, grate surface 444 square feet, heating surface 15,000 square feet. Fig. 2 shows a group of twelve Belleville generators, with same grate and heating surface. Fig. 1 shows a floor space of 1,713 square feet; fig. 2 shows a floor space of 1,401 square feet. Multiplying both areas by 14 feet height, we have: fig. 1 gives a capacity of 23,990 cubic feet; fig. 2 gives a capacity of 19,620 cubic feet. The difference in favor of Belleville generators is 4,370 cubic feet, or 97 tons of coal stowage.

For the comparison of weights, I estimate the weight of the battery of boilers in figure 1 at 362 gross tons, or 45.2 tons each, including water (26.6 tons material; 18.6 tons water.)

The group of Belleville generators, including water, weighs 193 gross tons, giving a reduction of weight of 169 tons; so that, while the space for coal stowage is increased by 97 tons, the saving in weight is 169 tons.

The weight of one Belleville generator is detailed thus—

Elements (1,250 square feet),	15,000 pounds.
Separator and purifier (steam drum),	1,100 "
Sediment chamber,	200 "
Feed water collector,	200 "
Water level column and feed regulator,	700 "
Piping,	400 "
Safety valve,	80 "
Stop valve,	150 "

Furnace doors,	200 pounds.
Grate bars,	1,100 "
Casing, uptake, ash-pans, etc.,	7,000 "
Cinders in casing,	1,900 "
Bricks,	5,000 "
Water,	2,400 "
	35,430 pounds.

Say, in round numbers $\left(\frac{36000 \times 12}{2240} =\right)$, 193 gross tons.

The comparison is made between a generator capable of sustaining 300 pounds steam pressure, with a strength of material to withstand 800 pounds hydrostatic pressure on the one hand, and a boiler built to carry 160 pounds, and for a hydrostatic pressure of 240 pounds on the other. The real comparative weights are 193 tons against 520 tons, were it possible to build the Scotch type for 300 pounds steam pressure.

This adoption of tubulous boilers is only a question of time. Already I read in the daily papers that the Ward system will be used on the *Coast Defense Vessel*, although this will only be a partial use and a partial success, because it is not allowed to place them in a desirable position on board owing to the presence of the Scotch type as a part of the boiler power. The Thornycroft boiler is also in use in the navy on the *Cushing* (see the last number of the JOURNAL, pp. 218–222), and is highly praised, thus bringing the Cowles boiler, its equal, into favor.

ASSISTANT ENGINEER F. C. BIEG, U. S. NAVY.—I would like to note a few points in the remarks of Mr. Coryell on the trial of the *Shearwater's* boiler and on the report of that trial.

In regard to the weight of the boiler, that given in the report was obtained, as stated, from the Atlantic Works, the builders of the hull and engine. It is the weight of the boiler complete in the ship, with smoke-pipe and casing, but without brick work, ash filling or water. Taking the estimated weight of smoke-pipe and casing from the 31,870 pounds given by the Board, the remainder would be about the same as the weight, 29,700, given by Mr. Coryell.

Owing to the lamentable want of reliable published data on the Belleville boiler, not many comparisons can be made, but the two cases given in the table may be of some assistance.

	Number of boilers.	One boiler.			Ratio, H. S. to G. S.	Weight in pounds per sq. ft. of—	
		Weight in pounds.	Heating surface, sq. ft.	Grate surface, sq. ft.		G. S.	H. S.
Mail steamer *Sindh*	12	29,607	921.25	24.57	37.49	1,205	32.14
French naval steamer *Alger*	24	38,147	926.	31.42	29.47	1,214	41.19
Shearwater, Coryell's figures*	1	40,600	824.	31.43	26.22	1,291	49.27
Shearwater, Board's figures	1	42,770	824.	34.17	24.11	1,251	51.90
Armored cruising monitor, Coryell's figures	12	35,430	1,250.	37.00	33.78	958	28.34

* Board's figures for brick work, ash filling and water.

I have no means of knowing exactly what is included in the first two weights, but perhaps Mr. Coryell will enlighten us and tell us exactly what the weights represent and if the data are correct. It will be noticed that the ratios in the first two cases are less than those given for the single boiler of the *Shearwater*, a circumstance which was noted in the conclusions of the Board which tried the *Shearwater*.

In regard to combustion, the concluding remarks of the Board state that "it is to be regretted that there were no means of increasing the rate of combustion and production of steam by a powerful forced draft." Had the boiler been forced, the moisture in the steam would undoubtedly have been increased. Mr. Coryell apparently misunderstands the object of this trial, which was not to show the capabilities of the Belleville boiler system under the most favorable conditions, but simply to test the particular Belleville boiler in the yacht *Shearwater* under certain conditions and limitations (the yacht not being the property of the Government), and to obtain some experimental figures of the only Belleville boiler then in this country, and further to supply, in this small degree, that want of "personal acquaintance" with the boiler which Mr. Coryell laments. I am sure that the Navy Department and the Bureau of Steam Engineering would have been only too glad to have had the opportunity to obtain additional

data and information in regard to the Belleville boiler during a recent competitive test of tubulous boilers, but, unfortunately, this boiler was not entered. Meanwhile, the few facts and figures that have been obtained from our own test and from the meagre published accounts, must supply the want which general statements, however numerous, fail to do.

DR. CHARLES E. EMERY, FORMERLY ASSISTANT ENGINEER U. S. NAVY.—The interesting paper of Assistant Engineer Leonard in the last number of the JOURNAL was so thoroughly discussed therein that at the present time it will simply be attempted to compare some curious differences in the proportions and general mode of operation of ordinary boilers in contrast with those of the tubulous type; to show that the economical performances of all boilers follow one general law, and to state some experiences that may prove of interest. The writer, from frictional contact, was originally impressed with the views of the older members of the profession in relation to the circulation and disengagement of steam in boilers, so ably set forth by Mr. Isherwood in the discussion, but extensive experience at a later period with the various types of sectional boiler showed that such views required modification. One of the most remarkable exceptions, not so novel in itself as in the form of statement, is ably explained in Mr. Thornycroft's paper, now accessible in a late volume of the Proc. Inst. C. E.

Our original conception of "convection" or "circulation" is exemplified in all boilers of ordinary type. Multiplication and various arrangements of the tubes make this circulation more and more active without changing its nature until, with the very small tubes referred to by Mr. Thornycroft, the action becomes violent and somewhat intermittent, like a geyser.

We then have this progression: a boiler in which the circulation is like that in a kettle, with steam and water rising at the center and water descending at the sides, will operate satisfactorily; so, also, special and sectional boilers provided with water up-takes and down-takes, from the heating surface to a separate drum, will circulate on the same principles and operate satisfactorily. Curiously, this will be the case whether the up-takes

be large or considerably contracted. Even some sectional-boiler makers in favor of free circulation acknowledge that the up-take can be made so large as to interfere with rapid circulation. We know that vertical boilers will operate well when there is a large space around the tubes for circulation; but the Naval launch boilers and Mr. Manning's modification of the same, where the shell is brought in close to the tubes till it acts like a corset to prevent free circulation, also operates well. So, also, a locomotive boiler, with plenty of room around the tubes, operates well, and it also operates well when there is very little room around the tubes; the fact being that, with a large area of down-take, a large quantity of water is moved at a slow velocity, while with less area a less quantity of water is moved, but at a higher velocity, produced by a greater head, due to the fact that less water is mixed with the steam during its upward movement and the density of the column is less. Some advocate restriction as a means of preventing foaming, as ordinarily understood, for the water about the heating surfaces is nearly all foam, and solid water cannot be lifted over into the engine. The extreme of this progression is a tube so long and narrow that, with solid water fed in to the bottom, the greater part of the tube will be a mass of foam, and mixed steam and water be discharged continuously or spasmodically at the upper end. It is moreover found that the steam and water of which the foam is composed can be separated in smaller space than is required with less vigorous ebullition. In other words, contrary to our old ideas of large steam space, large disengaging surface and quiet ebullition to prevent foaming, we can apparently obtain as good results in a boiler composed of long, narrow tubes, each of which foams vigorously, perhaps spasmodically, in true geyser style, though not foaming in the sense ordinarily understood where water is carried to the engine.

In ordinary boilers the steam passes upward and bubbles through the water at the disengaging surface, which plan operates satisfactorily, but with the geyser type of boilers there are differences of opinion whether or not *it is* best to discharge the upward current of mixed steam and water under the surface of the water in the drum or entirely above it. Mr. Thornycroft

advocates the latter, and this system is adopted with modifications in the Ward and Belleville boilers. A gentleman discussing Mr. Thornycroft's paper claims, however, that it is better to discharge the water and steam from small tubes below the water level in separating drum. It may still be considered doubtful which system will carry least water to the steam pipe. A similar question arose years ago in regard to separating from the air delivered the water used to fill the clearances of an air compressor. Naturally it was first thought best to deliver *above* the surface of the water in a separating tank, but somebody dipped his pipe below, and considered that he *washed* the particles of moisture from the air by that method of procedure. It will be better to let the promoters of special designs urge that a particular method of operation is better than another, and for engineers to accept all as having merit, and to carefully study the principles which make it possible for entirely dissimilar methods to produce substantially the same results. In the end it will probably be found that each mode of operation is adapted to a particular set of conditions.

The Herreshoff, Belleville and Thornycroft boilers are strictly tubulous boilers and all operate upon the geyser system, the difference being more in mechanical details, and the Herreshoff being somewhat distinctive from the others in the mode of operation, principally from the fact that the water is fed into the coils with a pump rather than circulated by difference of density and that the current is down instead of up. The Belleville has the typical geyser operation, and the delivery is through special spouts above the water in the drum. Its chief distinction is a mechanical one, in that well-fitted heads forming return bends are grouped to form coils, but so that any section may be removed and both the insides and outsides of the tubes be properly cleaned. The better construction materially increases the weight compared with other forms. Ward has grouped his pipes in a way that sections can be taken out vertically and put in order, and has succeeded marvelously in the use of joints which *a priori* would seem quite insufficient for the purpose. He unites several long tubes into vertical headers, and these again into

one main discharge, which is above the water line. Thornycroft has boldly jumped past all details designed to make the tubes accessible for cleaning inside and all customary modes for cleaning outside, and made up the boiler simply with reservoirs and long, crooked tubes, so elastic as to avoid straining at the joints. The step is a bold one, but it has saved so much weight for a given power that success has long since been assured for boilers not regularly in use. Few will be bold enough to predict, however, that such a boiler is adapted for continual use in vessels, for instance, of the merchant service or those of the Navy that make long cruises away from a base of supplies. It would seem that all of these boilers are to a greater or less extent temporary expedients, none the less necessary for particular uses, and that each is liable to difficulties peculiarly its own. The introduction of each must be made to a large extent by guaranteeing orders. Mr. Ward has been pluckily doing this for a sufficient period to enable us to know what he can do, and the difficulties he has developed do not seem to be insurmountable. The Belleville boiler, with its very excellent construction but comparatively great weight, compared with other pipe boilers, needs to be introduced more largely in the United States on vessels that have hard work to do continuously, and where it must be operated by the same men that operate machinery of more customary type.

One of the fast yachts owned in New York had originally two of the Herreshoff boilers with circular coils, which failed and were replaced with two similar boilers, which also failed, from the fact that the descending water was kept, by centrifugal action, away from the sides of the pipes exposed to the fire, permitting them to become very hot, finally compressing the metal beyond the limit of elasticity so that the *pipes cracked transversely on the inside of the coils when the boilers were cooled.* The writer was consulted in relation to the substitution of Herreshoff boilers of the new flat-coil return-bend type. The so-called "wrought-iron" bends were tested and found to be cast steel, which would perceptibly harden on being heated and cooled. The owner was anxious to put in *one* large boiler of this type, thereby saving

sufficient room to enable him to locate his galley below, where it could communicate with a main-deck dining room by a dumb-waiter. The season was rapidly approaching, and the writer finally assented to having the type tried, if *two* boilers were used, separately connected. The owner tried by argument to have the decision changed, but was finally told that it was known he would go out in a heavy sea for the excitement, and the writer was not willing to risk his (the owner's) wife and children in that way with a new boiler practically untried on such a scale. The owner sensibly but reluctantly gave his consent; two boilers were put in the boat and ran nearly the whole season without a particle of difficulty, of which, it is proper to say, the writer was frequently reminded. Finally, no less a person than the *President of the United States* was taken on board. After the vessel left the dock, the water in one boiler went up the smoke pipe through a broken fitting, considerable steam from the other boiler following, and sufficient escaped into the fire room to frighten the firemen, who hastened on deck. The captain, who was also pilot, with apparent awkwardness, got his vessel behind a slow tug; the engineer shut off the disabled boiler and reassured the firemen; the blower was pushed and in a few minutes the yacht shot ahead like an arrow, and proceeded to her destination without even the owner knowing that anything had happened, until the President was landed. While the stock of such men lasts the manufacturers of special boilers are safe, but what if the consulting engineer had allowed the owner to risk one boiler of that type?

The next year the same owner had become impressed with the Ward boiler. His own judgment was good, and before calling he had already arranged for a larger grate than proposed. To be frank, the writer could not at that time understand how the boiler could operate without lifting water dangerously, with such small steam space and small disengaging surface, but the evidence adduced showed that it *did* operate satisfactorily, and he simply recommended more heating surface to make the work as easy as possible, remarking that the boiler ought to last longer than the others that had been tried, and that, as there always was a supply

of water in the drum to feed the tubes without depending upon a pump every moment, he considered that there would be little danger of the boiler being disabled suddenly. One boiler was accordingly permitted this time, and the desirable arrangement of dining room and galley carried out. It developed that less expert firemen were required, and the boiler gave excellent satisfaction, in consequence of which another boiler of the same size was put in the U. S. Revenue steamer *Manhattan*. In due time the drum of this boiler "bulged" just above the fire. Mr. Ward put in a seamless one at his own expense, which in due time also bulged; but, after cleaning the boiler thoroughly to remove the grease, no further difficulty was experienced, and it is believed that by putting in a circulating tube, so as to separate the upward current due to the heating surface of the drum, from the downward current required to feed the tubes, as much grease can be used in the engines as is practicable with other boilers. About this time the yachtsman was addressed to see if he had had any difficulty, and he wrote a letter extolling the boiler in the highest degree. It developed incidentally that he had not escaped without some trouble, but the boiler had done so much better than the previous ones that he thought his difficulties not worth mentioning.

The economy of a boiler does not depend upon its type, or the particular way the water is circulated, but upon the simple principle enunciated by the writer in his paper on "Boiler Proportions," in the general report of the judges of Group XX, Centennial Exhibition, substantially embodied in a lecture at Sibley College. (See *Scientific American, Supplement*, Nos. 687–688, in which the cuts, however, are too small to study.) The principle is simply that when there is proper circulation of both the water and the products of combustion, the economic result is a function of the average quantity of combustible burned per square foot of heating surface. It is important that there be proper circulation, not only of the water, but of the products of combustion. Very many special boilers have large chambers and curious shaped passages, so arranged that the products of combustion do not necessarily pass over all portions

of the heating surface; the current takes the lines of least resistance, and while the surface actually passed over is very efficient, the average efficiency is low. On examining the paper referred to it will be found that for low rates of combustion the principal results were obtained from the experiments made at the Brooklyn Navy Yard during the war. The equation and resulting curve given, average the varying results of different experiments and show the law of variation. The law is expressed (1) $E = \frac{46.045}{c + 3.016} - 1.067$, in which E equals the number of units of evaporation (pounds evaporated from and at 212°), and c equals the number of pounds of combustible consumed per square foot of water heating surface per hour.

The results obtained in New York with a Martin boiler with grate reduced so as to give a large proportion of heating surface to grate were, as stated in the reference, the highest which can probably be obtained in practice, as the coal was of excellent quality and the firemen and engineering force had become unusually expert. Some experiments plot higher than the curve, but others plot below, and the position of the curve was such as to reach a limiting point based on the theoretical evaporative efficiency and practically average the various experiments. The results with the Thornycroft boiler, supposed by some to be higher than possible, will, on comparison with the curve, be found lower than has actually been obtained by experiments in this country. The result by curve corresponding to the low rate of combustion is 13.74 units of evaporation; the results with the Thornycroft boiler were only 13.4 units. The comparison for another experiment is 12.44 to 12. These are very excellent results compared with those given by other boilers referred to in the *Scientific American Supplement.* It is interesting to compare the report of Prof. Kennedy's experiments with the paper of the writer, for the reason that he adopted the same method of treatment. The paper of the writer, however, assumed that the coal had an ultimate average of efficiency of 15 pounds from and at 212°, whereas Professor Kennedy found experimentally that the coal (Welsh probably) used in the trial of the Thornycroft boiler

had a calorific value of 15.4 units of evaporation. This only affects what is called the "ultimate efficiency" in the writer's paper, but not the comparison where the corresponding ordinate of the curve is made the denominator, when the result is called the "practical efficiency."

It being settled that the economy of the different types of boiler is based on the same law, the efficiency is frequently very low, which is due generally to the improper distribution of the heated gases over the heating surfaces, whereby a large portion of the gases can take a short circuit to the stack. This difficulty is easily overcome in ordinary boilers by reducing the cross area for draft, so that the whole heating surface becomes efficient, which can be done if the products of combustion either pass through fire tubes or between water tubes. With tubulous boilers it is more difficult, as all possibility of direct access must be given up if the tubes are massed closely together in a flue. In the writer's opinion, the best form of boiler for reasonable rates of combustion, is one with inclined tubes connected by up-takes and down-takes to a chamber or drum above, as in many sectional boilers. This exact construction with a water-wall fire box was used many years ago in what were called "Little Giant" boilers. Dickerson attempted to improve them by putting superheating tubes through the upper chamber, thereby producing mechanical failure in the boiler of the *Algonquin*. The unimproved boilers have, however, always given good results. Of two, put in small launches for the U. S. R. M. in 1860, one is still running in the harbor of Baltimore, and two years ago, when the engineer was ordered to cut hand holes and examine it thoroughly, he could not recommend any repairs. The water supply had been from the city mains. In the other, stationed at Savannah, the boiler was renewed four or five years ago, but no improvement was attempted; and the third boiler of this type has been put in place of a vertical boiler without taking any more floor room, and but slightly encroaching on other quarters at two of the upper angles. The boiler of Mr. Towne is on the same general principle. The crossing of the tubes causes circulation at both ends of the boiler, and the trial shows economi-

cal operation. This general form of boiler operates so nearly like what may be called ordinary boilers that it will be more popular than a strictly tubulous one, and by using very small tubes and massing the same closely the weight per horse power can be satisfactorily reduced.

It is necessary in testing any form of boiler to ascertain the quality of the steam. For this purpose the ordinary barrel calorimeter is believed to be sufficiently accurate. When the percentage of moisture is very small the errors of observation are such that the experiments cannot be repeated with minute accuracy, but the average of several experiments eliminates the errors on a well known principle, even though one or more of the experiments may show superheating when there is no superheating surface. It is the error of observation that shows the superheating, but this error is as liable to be the other way next time, so the average is the true result. In the above article on Boiler Proportions (in Report of Judges, not in *Scientific American Supplement*,) this subject is discussed, and it is there shown that the steam should be taken from the main steam pipe at a very low velocity, and that the contraction necessary to reduce the quantity delivered be made at a distance from the main steam pipe, to prevent superheating of the steam for calorimeter by that passing on to the engine.

It may be added also, that the quality of the steam should be tested from the steam pipe, and not from a portion of the drum, where the steam may be quiescent and much drier. As suggested by Mr. Loring, the weight of the water in the barrel should in all cases be the same, the reason being a simple algebraic one, as then the weights for different experiments can be added together and the final results worked out from the sum. Sometimes incorrect results are secured by blowing the water into the barrel too fast, as the demand on the boiler is sufficient for the moment to produce priming.

If our time is not up, we will recall, *apropos* of Mr. Loring's "device of the devil," viz.: automatic apparatus, the story of the late lamented Henry R. Worthington, about the steam canal boat of his younger days, its vertical boiler and its automatic

feed, which latter worked so well that he customarily spent his watch on deck, except when it was time to fire up. He would give a description of the time when the groaning of the engine from the use of superheated steam called him below, to find the water entirely out of sight and everything nearly red hot. *That* boat could be tied up to the bank, and when this was done the overalls of a workman, sent to do a trifling job in the boiler, were found in the automatic valve of the small steam pipe supplying the steam pump. Mr. Worthington never used an automatic apparatus afterward. On telling this story to a Russian associate of the writer on a group of judges at the Centennial Exhibition, who wished an award for one of his countrymen for a similar device, he finally grasped the point and expressed it, "My dear Mr. Ameree, you vish me to understand zat ze better ze device work, ze more dangerous it become", which explanation was acquiesced in.

PASSED ASSISTANT ENGINEER BAIRD.—I feel disappointed that the Herreshoffs have not availed themselves of our invitation to take part in this discussion, for there are admirable points in their boiler which should be recorded.

My first acquaintance with the Herreshoff boiler dates back about seventeen years, when they made their proposal to the Chief of the Bureau of Ordnance to build the *Lightning*, and to guarantee an unprecedented speed. The Chief of that Bureau (Commodore Jeffers) asked me to examine all the drawings, descriptions and proposal, and give him an opinion. The proposed boiler was the original coil boiler of the Herreshoffs; it was fed at the top, and discharged into a separator from its lower end. The scantlings of the hull and the engine, though apparently too light, were found to be sufficient when I calculated the strains. I was not, however, quite ready to endorse the boiler, so recommended that a smaller Herreshoff boiler be purchased and fitted to an English-built launch at Newport, for test. This was done, and it proved successful, as also did the *Lightning*. The unprecedented speed of this little craft not only made the Herreshoffs famous, but called the attention of engineers to the possibilities of tubulous boilers.

My next experience was with the steam yacht *Fanny Hayes*. She had a Herreshoff boiler, which was continuously "blowing out." At the request of the owner of this little yacht, I made an examination and a test to ascertain the trouble, which was found not in the boiler, but in the feed pump. This was a single-acting plunger pump, worked vertically from the cross-head. Its great number of strokes per minute, at each of which its valves had to open and close, and the water in the barrel and chamber had to change direction, together with the dirty water which came from the keel condenser, were quite sufficient to impair the efficiency of the pump, to produce a scarcity of water, and a consequent "superheat" of the coil. My remedy was to put in a donkey feed pump, which, at 120 feet piston speed, fed the boiler, and from that time forth there was no trouble with "superheats" on board that yacht.

In 1881 we purchased two Herreshoff steam cutters for the Fish Commission Steamer *Albatross*, which vessel was then being built under my superintendence. The original form of Herreshoff boiler was used in these cutters. The feed pumps were worked from the cross-heads, and there was no other means of feeding the boilers. There were small fresh-water tanks in these boats for supplying the leaks. Clean distilled water was used in the boilers, and the feed pumps and tanks were cleaned at every opportunity. The grate bars were replaced once in every eighteen months; new casings and smoke pipes were necessary at the end of two years; the only parts of the original boilers left at the end of two years, were the coils, separators and a few fittings. Twice a year I had the boats landed, took out both engines and boilers, replacing every defective part of the boilers, and re-bushing the engine journals and re-lining the engines at these times. The boilers (the coils) by this kind of attention lasted six years.

The boats did a great deal of running, and generally well up to full power. We sometimes carried them at the davits for forty-eight hours at a time with the fires banked. As soon as fires were hauled the coils were swept and the ash-pit and furnace closed and smoke pipe covered, to keep out moisture. The com-

manding officer of the *Albatross*, as well as the Commissioner of Fisheries, never once restricted me in any repairs or changes I desired to make in these boats, which enabled me to keep them in good condition. My experience has, therefore, been confined to the original and not the "improved" Herreshoff boiler.

The one great difficulty I have found with this type of boiler is to find or to train men to run it. As all others are "water-level" boilers, and as the height of water is the thing most carefully watched, it naturally makes a man nervous to confront so radical a change. This original Herreshoff boiler contains the greatest strength for a given weight of metal. It has a positive circulation; there are no pockets to deflect the water; it takes its one unalterable course from its receiving point to the separator. It contains a relatively large furnace, that the *rate* of combustion per square foot of grate may be diminished, with a given total combustion.

The steam, I believe, is mostly formed in the separator, where the pressure is somewhat diminished; and if this be true, the coils are run pretty full of water, the weight of which in this type of boiler is nearly as much as in other tubulous boilers.

It will not be out of place to mention here some experiments made by Chief Engineer Thom Williamson on the flow of globules of steam through inclined tubes.

These can be illustrated by taking a glass water gauge full of water and admitting a bubble of air at the lower end; if the tube is vertical, the bubble of air (or steam) will so wobble in its course to the surface as to require greatly more time than if the tube be inclined. Mr. Williamson repeated this with the tube at different inclinations, and found the velocity of the bubble to reach its maximum when the tube was at 45 degrees from the horizontal; he found, also, that the difference was not great between 15 and 45 degrees, but when the tubes were inclined at a lesser angle than 15 degrees, the retardation of the bubble became much greater. This experiment was made with tubes of a given length, and the velocity of the bubble was actual, and not measured as to vertical height.

The rapid natural circulation in inclined tubes seems to ex-

plain the dryness of the steam in the admirable little boiler designed by Mr. Towne for the steam launches, in which the tubes, I believe, are placed at an inclination of about 16 degrees.

CHIEF ENGINEER CHARLES H. LORING, U. S. NAVY.—By some misunderstanding two errors that significantly obscure the meaning crept into the printing of my remarks on the paper under discussion. On page 195, Vol. 2, of the JOURNAL in line 17, "pound of coal" should read I.H.P., the reference being to the table on page 165, from which this quantity is easily obtainable. On page 197, lines 8 and 9 should read "original weight of the barrel be used, in the computations of each test in a continuous series." In reference to the subject of these two lines, I wish, with the Council's consent, to say that the variations in this weight from the original during an unbroken series of tests can only be caused by the presence of water,—either from imperfect drainage, or by absorption. This water is necessarily heated along with that run in, and is therefore a part of W and not of *b*. The discussion of a not very important matter needs no apology here, I trust, involving, as it does, a physical fact.

Regarding the proposition of the first lines of the same paragraph, it is submitted, that if supersaturated steam escapes through two pipes, the condition attending each must be similar to ensure similar hydrous character in the discharge from each.

Velocity of flow at the place of divergence is one of these conditions, and it is certain that this cannot be equal in both, if one of them has a largely reduced internal cross-section in any part of its length, beyond the place of separation. It is beyond doubt that, in any usual arrangement of calorimeter pipe, the conditions are such that a less percentage of water passes through it than goes with the steam through the main pipe. This fault is, perhaps, unavoidable, but an unobstructed pipe is easy to get.

Mr. Coryell's elaborate description of the Belleville boiler is exceedingly interesting. In the definiteness of its detail it places the principles and peculiarities of the system very clearly before the readers of the JOURNAL.

Our debt to him would have been larger had he also given us some reliable data of performance upon which beliefs could be

based, and comparisons of its efficiency made with other boilers of its type. Such data might dispose of some questionings as to the possibilities of certain parts of the system, as, for instance, the limit of potential evaporation under the alimentation by gravity. Will this system of alimentation serve for an evaporation due, say, to the consumption of one and a half pounds of coal to a square foot of H.S.? Again, some data is required to show how much of the water from the steam condensed by contact with the feed water in the drum can be evaporated, in practice, by the heat units liberated in the sudden dropping of the boiler pressure to that of the engine pressure.

Satisfactory answers to these questions would add largely to the repute of the boiler in the minds of professional men, whom experience has taught the vanity of relying on vague generalizations made by unknown brethren.

Many vain attempts have been made to use for marine purposes the locomotive boiler designed with the proportions usual in such boilers intended for the rail. That it does so well in its usual place, and so very badly afloat, is not easy of explanation in an altogether satisfactory way. The difference of performance must of necessity be from difference in the conditions attending its use. Of these I can see but one at all likely to produce such unlike results. When the boiler of a locomotive is passing rapidly along the rails, it is subjected to a succession of tremors, quite capable of dislodging the particles of steam from the heating surface immediately they take form, ensuring almost constant water contact with this surface. If this be true, the overheating of the tube plate and tube ends is prevented, which appears to me to be the best explanation why a locomotive boiler endures so well the stress of large coal consumption. The absence of priming in a boiler having all the conditions of a priming boiler may be from the inability of the steam to accumulate below the water line, by adhesion, and rise to the surface intermittently. I see no reason for supposing that of two locomotive boilers, alike in all respects except the length of tubes, and consequently in area of H.S., one should not endure as well under high rates of combustion as the other. Or if there is any

difference, it should be in favor of the shorter one, since the total expansion of its tubes would be less than in the other.

Mr. Asa M. Mattice, Member.—I have read with great pleasure the paper on this subject by Assistant Engineer Leonard, and the discussion thereof. Until recently but little has been generally known on this subject, nearly all that has appeared in print being prejudiced in favor of some one type or make; but this paper sets forth in little space an exposition of nearly all the tubulous boilers that are adapted to naval uses. I have not the slightest doubt that the tubulous boiler is the boiler of the future for war ships, if not for fast vessels of the merchant marine.

In boilers for naval vessels, however, too much attention must not be paid to heat efficiency to the neglect of other very important points. The boiler must be easily repaired, and that by the most ordinary mechanics; the separate parts must be of the smallest number and simplest description; and, above all, all parts must be easily accessible for repairs and cleaning without the removal of surrounding bulkheads and parts of the boilers themselves. Great care must be exercised in the building in order to make the life of the boiler as long as possible. The principal difficulty that has been experienced seems to be fitting of the tubes. This can be minimized, if not avoided altogether, by a rigid inspection of the tubes and a careful pickling to remove all black oxide scale, all tubes being rejected which are not perfectly smooth inside, and free from scale after pickling. Or, in lieu of pickling, I would suggest that the tubes might be rumbled until bright, an iron rod being put in each tube before being placed in the rumble, so as to clean the inside. This could be done, of course, only for boilers having straight tubes. I think that a little experience will prove that the sheet-iron cases which tubulous boiler makers use will have to be thickened. Few people except sea-going engineers are aware of the very rapid corrosion of thin plates under the combined action of heat and salt air. Care must also be taken that not the slightest greasy deposit may collect in the tubes and cause overheating. Every dollar spent for potash for periodical "boiling-out" will save a good many dollars' worth of boiler.

With some points brought up in the discussion I must beg leave to differ. On page 187 the statement is made that the admission of cold air has no influence on the leakage of the tubes. I have had too much practical experience to the contrary to agree to this. Many boilers with very "tender" tubes will run a long time without giving any trouble until sometime the furnace doors are left open too long, after hard firing, and then there is a job for the boiler maker's gang. The programme is generally about as follows: engine stopped suddenly to take soundings without any warning having been given; safety valve pops; officer of the deck orders to "stop blowing off;" jam down safety valve and open all doors; shower-bath begins in the back connection; ship laid up in the next port while tubes are being re-expanded, ferruled, or renewed, &c., &c. The higher the boiler pressure, the greater the trouble, on account of the thick tube sheets. When the temperature is gradually increased or diminished, the tube sheet and the tube expand and contract together; but, when a blast of cold air comes along, the thin tube end cools faster than the sheet, contracting the diameter of the former, making it a loose fit in the hole, and allowing the tube sheet to bulge ever so little, but this slight movement does the damage. It is known that "tender" tubes can be protected to a great extent by ferrules which are not even close fitting, as long as they cover the end of the tube and its interior as far as a little beyond the inner edge of the tube sheet. As to the use of brass tubes in naval boilers, mentioned on page 192, I would call attention to the fact that the English have used iron and steel tubes almost, if not quite, exclusively for many years, and all of the boilers for our new vessels have been and are being built with tubes of the same material, that is, for all vessels begun within the last half-dozen years.

The statement made on page 186, regarding the heat-conducting power as being inversely proportional to the absolute temperature, is important if based on actual observations. It would be interesting to know on what authority the statement is made, and where further data may be obtained.

Regarding the question raised by Chief Engineer Loring

about the choking of a calorimeter pipe, I think that the reduction can make no appreciable difference, provided that the proportions of all parts are such that steam enters the calorimeter pipe and travels for a short distance in it at the same rate at which it flows in the main steam pipe from which it is taken; also that the calorimeter pipe is thoroughly filted; and, of course, provided that in case of a small boiler the steam for the calorimeter is not taken so fast as to crowd the evaporation. The reduction of area will, of course, cause superheating; but if the pipe is well covered with a good non-conductor, and is thoroughly blown through before steam is turned into the barrel, all the heat of the steam eventually finds its way into the water.

It is to be regretted that the Belleville boiler people would not submit a boiler for the competitive test instituted by the Bureau of Steam Engineering. The only reliable data that are available concerning this boiler are from the test on board the *Shearwater*, where, unfortunately, a forced-draft test could not be made. I had hoped to see some data presented in the discussion of Mr. Leonard's paper, but was disappointed. It would certainly seem to be to the best interests of the makers of the boiler to advance some reliable data to substantiate the high claims that they make. I can see no reason why the boiler should not give a good account of itself, as the design appears to be good, and it has been in use long enough for faults to be discovered and corrected.

Regarding the reported phenomenal evaporation of 13.4 pounds of water from and at 212°, per pound of coal, by the Thornycroft boiler, it should be borne in mind that whereas, at high rates of combustion the upper parts of the tubes of this boiler are filled with a mixture of steam and water, or perhaps with dry steam only, with light fires the tubes are probably completely filled with water, thus giving a much larger water-heating surface. Besides, in the test made by Prof. Kennedy, the grate was bricked off so that the ratio of heating surface to grate was about 70 to 1. However, as Mr. Griffin has observed in this discussion, experience has shown that some tubulous boilers

carry more water over with the steam at low rates of combustion than at high ones, and it is reasonable to suppose that such was the case with the Thornycroft boiler in question. Again, we must take into account the fact that the test was made with a "running start" and a similar finish, the state of the fire at beginning and finish being "judged;" and that the test was of only five hours' duration. A test made in this way cannot be expected to be as accurate as one where steam is raised, fires hauled, and the fire for the test started on clean grates, then the fires hauled at end of test and the residue weighed. The "ashes," or rather all that fell through the grate, about 233 pounds in all, and which was shown by analysis of the fuel to contain about 22 pounds of incombustible, was put back on the fires during the last hour of the test. I was about to show how this would affect the "judging" of the fires, but upon looking over the discussion of Mr. Thornycroft's paper in the Proceedings of the Institution of Civil Engineers (Vol. XCIX), I find that Prof. Unwin had already done the same thing; so I can do no better than to quote from the report of his remarks:

"The weight of coal, measured out on the stoke-hold floor and put into the furnace, was 1,006.5 pounds, and it was taken to be the consumption of the boiler in the five hours. Obviously, that involved a considerable assumption. The trial began with fire in active combustion, and ended with the fire in active combustion, and, in order to be sure that only 1,006.5 pounds of coal had been used, it ought to be certain that the fire was in exactly the same condition at the beginning and at the end of the trial. He supposed that, with the 26 square feet of grate, there would be at least 250 pounds of coal on the grate in the normal condition of working; therefore, a complication was introduced by the 1,006.5 pounds of coal having mixed up with it, at the beginning and the end of the trial, a quantity of 250 pounds, or 20 per cent., which had been absolutely guessed—not measured at all. In one respect Professor Kennedy's trial differed from any that he had known. During the last hour the whole of the ashes had been put back on the fire. It appeared, from the analysis of the coal, that there must have been produced out

of the 1,006.5 pounds, 22 pounds of incombustible ash. There was, therefore, in the fire at the end of the trial an amount of 22 pounds of incombustible ash which was absent at the beginning, and as the fire could only be judged by its bulk, he thought that Professor Kennedy, even on his own mode of estimation, ought to have added 22 pounds to the 1,006.5 pounds as the coal burnt on the trial. That was, a quantity of 22 pounds of incombustible matter was on the grate at the end of the trial, which replaced 22 pounds of combustible on the grate at the beginning. It might be said that 22 pounds was a small quantity, yet it was 2 per cent. of the whole amount burnt. But he ventured to say more than that; Professor Kennedy understood well that the weak point of the trial was the judging of the fire at the beginning and at the end, and he had made a remark about it, stating that in the diagram of coal consumption during the trial, the straightness of the line would form a criterion whether the fire was in the same condition at the beginning and at the end of the trial. It would be seen from the diagrams that towards the end of the trial less coal was put on the fire. The line was made tolerably straight by adding the weight of ashes, containing only 85 per cent. of carbon, and treating it as if it were good coal. To some extent that was right; but if the diagram were appealed to, the fair inference from it would be that the fire was let down at the end and was not in the same condition at the end as it was at the beginning. As the fire could only be judged by bulk, and the bulk was a bad criterion of the weight (for the coal might burn half away without showing much difference in the bulk), he thought an error of at least another 42 pounds might have been made. The possibility was, therefore, that there was another error of 4 per cent. in the judgment of the fire. On the whole, allowing 2 per cent. for the ashes put back, and another 4 or 5 per cent. for the probable difference of the fire, it was quite possible, the trial lasting only five hours, though made with the greatest care, might be out by 10 per cent."

Unfortunately, this is not the first recorded performance of a boiler trial where the result depends largely upon the skill of the fireman in jockeying the fire into a condition to be judged to the

best interests of the boiler maker. Engineering literature is full of accounts of such trials, but the absence of details often prevents the separation of these fairy tales from reliable data. However, I think that Professor Unwin's estimate of a possible error of 10 per cent. may be a little high. As far as combustion is concerned, the boiler ought to have shown a high efficiency, for it was practically perfect. Of five analyses of the chimney gases only one showed any carbonic oxide, and then only 0.5 per cent., or an average of 0.1 per cent. for the whole trial.

ASSISTANT ENGINEER S. H. LEONARD, U. S. NAVY.—In the last number of the JOURNAL I made the following statement: "It is thus seen that this boiler (Belleville) has no practical advantage over the Scotch boiler, either in weight or space occupied." This statement was based upon the figures, as shown in the table, which were the most reliable at hand. It is certainly unfortunate that the only experimental data of the Belleville boiler made public is that of its test aboard the yacht *Shearwater*. For some unexplained reason it now appears from Mr. Coryell's statement that the designed values of this boiler were all wrong, and that it was abnormally heavy and bulky.

Mr. Coryell gives as the weight of a Belleville boiler (37 square feet grate and 1,250 square feet heating surface) 36,000 pounds, or 28.8 pounds per square foot of heating surface. The weight per square foot of heating surface in a battery of ten Belleville boilers fitted in the French war ship *Hirondelle* is 31.5 pounds; in the *Milan* (12 boilers) 32.1 pounds; in the *Voltigeur* (6 boilers), 33.4 pounds.

If we assume 32 pounds per square foot of heating surface in the twelve Belleville boilers given by Mr. Coryell in his illustration, the total weight will be *214 tons*, instead of 193 tons.

In his comparison Mr. Coryell is at fault in changing the dimensions of the boiler space to suit his boilers. Had this been permissible, double-ended Scotch boilers could have been used, thus saving space and weight. The weight per square foot of heating surface in the single-ended Scotch boiler is 53.5 pounds, while in the double-ended boiler it drops to 41.5 pounds, or the

same as a single-ended boiler designed for about 80 pounds pressure per square inch.

The height charged to the Scotch boilers in Mr. Coryell's arrangement should have been 12 feet 8 inches, instead of 14 feet, as it was *not necessary* to place the stop valves as located.

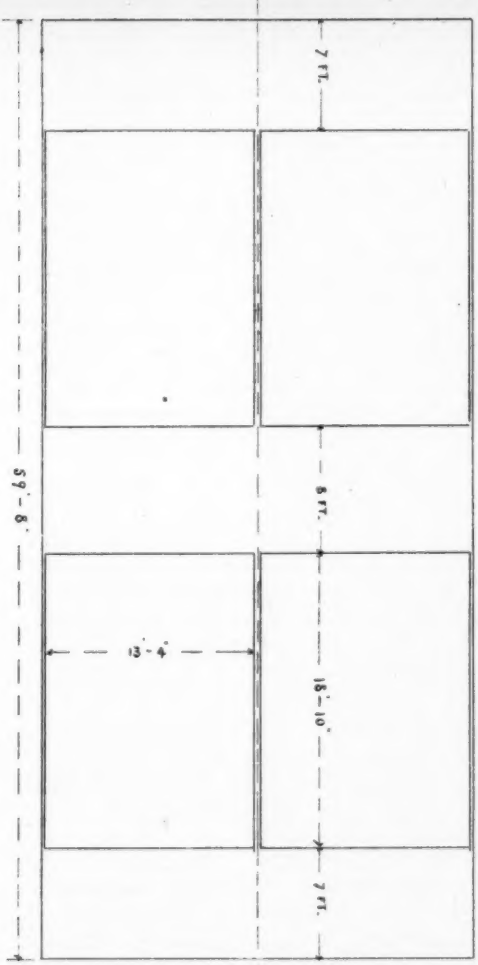
Assuming that the dimensions of the boiler space could be suited to the boilers, I have given arrangements for the Scotch, Thornycroft, Ward and Belleville boilers for 450 square feet of grate area, allowing the heating surface to remain as designed in each boiler, in order to arrive at accurate results as to weight and space of each type. It will be noticed that the heating surface in the Thornycroft and Ward boilers is considerably in excess of the others, and for which no credit is made. In the arrangement of the Belleville boilers I have taken the liberty to increase Mr. Coryell's width of fire room to 7 feet, for even if the grate is but 5 feet long (as recommended in the text), the above width is by no means too great to properly work the fires. Fire rooms of 7 feet and 8 feet are allowed in the arrangement of Scotch boilers, whose grates are but 5 feet 3 inches long. In the other arrangements a slightly better proportion of room is allowed. From these arrangements it appears that the saving of space of the Belleville over the Scotch is 706 cubic feet, or equivalent to about 16 tons of coal.

The saving in total weight is 101 tons.

The saving of space by the Ward and Thornycroft boilers over the Scotch is respectively 4,270 cubic feet and 4,498 cubic feet, or over 100 tons of coal. The saving in weight is respectively 199 tons and 184 tons. We thus see that the weight of the battery of Belleville boilers is approximately twice that of either the Ward or Thornycroft.

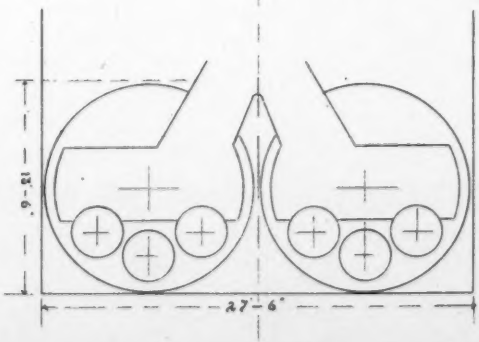
The weight of each set of boilers, empty, is as follows: Belleville, 180 tons; Scotch, 174 tons; Thornycroft, 90 tons; Ward, 82 tons.

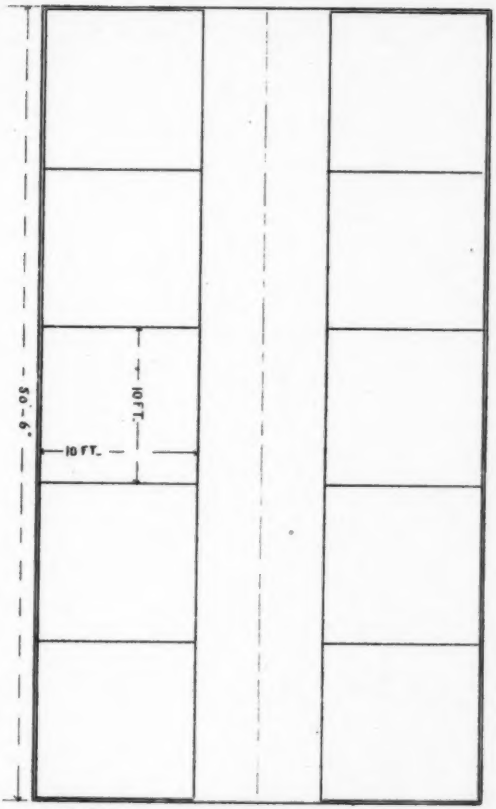
I think Mr. Coryell assumes too much when he states that the Belleville boiler in the table (last number of the JOURNAL) is compared with tubulous boilers of such thin material as to sacrifice durability and safety. The tubulous boilers quoted have



GRATE SURFACE	446	SQ FT.
HEATING "	13034	" "
FLOOR SPACE	1639	" "
TOTAL "	22126	CU. "
WEIGHT	399	TONS

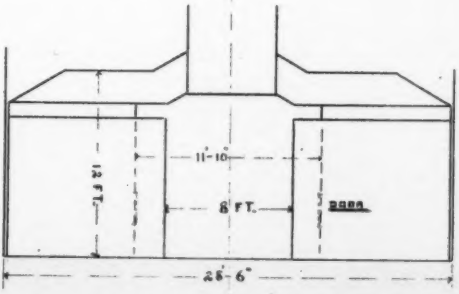
SCOTCH

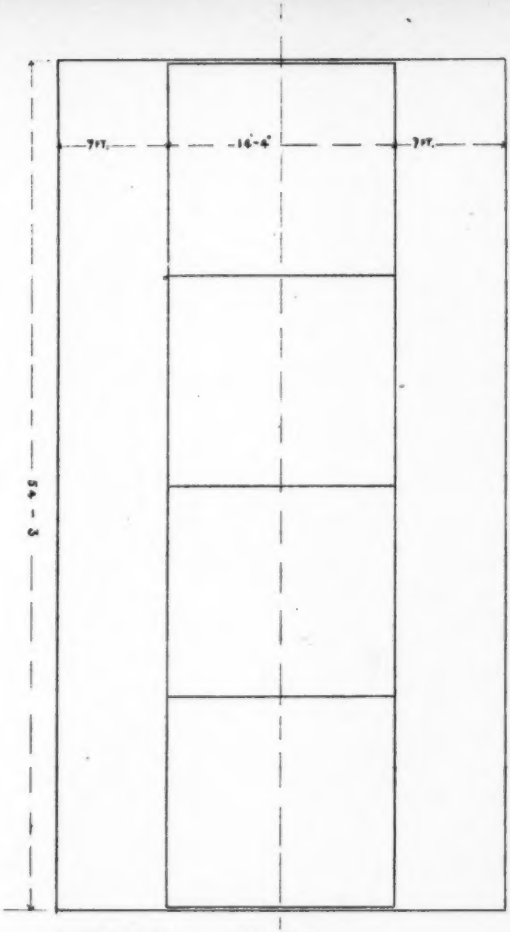




GRATE SURFACE	496 SQ.FT.
HEATING "	23738 " "
FLOOR SPACE	1437 " "
TOTAL "	17628 CU "
WEIGHT	110 TONS.

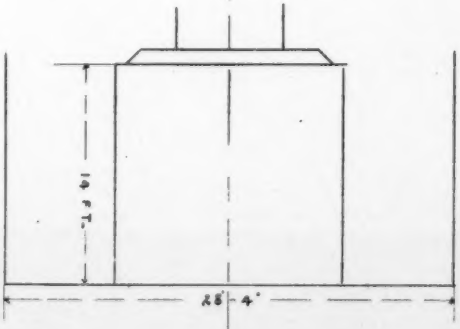
THORNYCROFT.

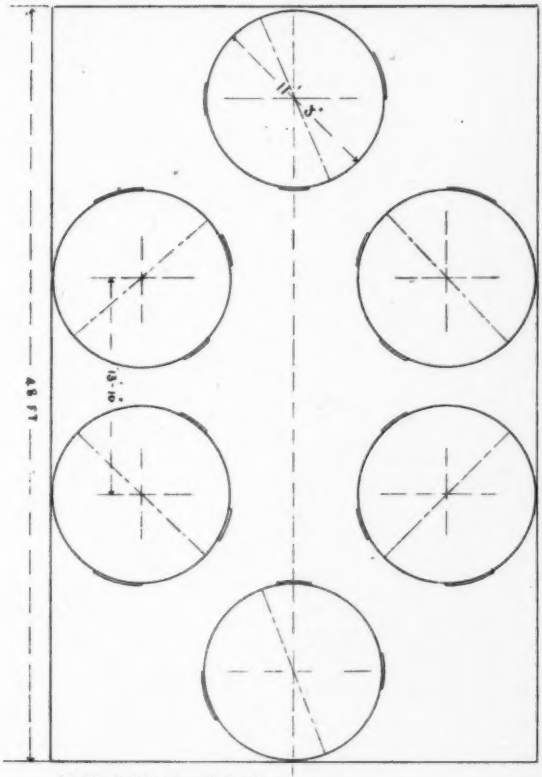




SKATE SURFACE	444 SQ.FT.
HEATING "	18000 " "
FLOOR SPACE	1830 " "
TOTAL "	21420 CU. "
WEIGHT	193 TONS

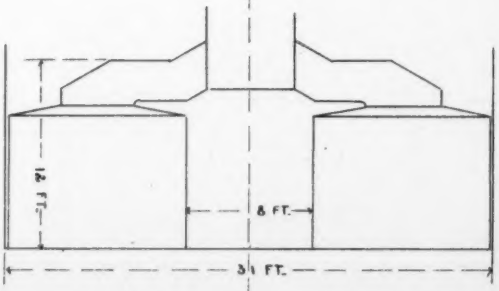
BELLEVILLE





GRATE SURFACE	650 SQ. FT.	
HEATING "	17625 " "	
FLOOR SPACE	1665 " "	
TOTAL "	17865 CU. "	
WEIGHT	95 TONS	

WARD-



been in daily use for the past three to ten years and have shown no special lack of safety or durability. Some of them are tested and worked at the same pressures as the Belleville. What the average life of each type of boiler is under average conditions has yet to be ascertained, and the same remark can be truthfully made of the Belleville type.

Until 300 pounds steam pressure becomes commercially useful, I fail to see the merit in a boiler built to withstand such pressure, thereby increasing its weight, cost and space occupied.

If I understand Mr. Coryell, the cause of increased weight of the Belleville boiler is the use of increased scantlings over and above those required for a certain working pressure, to offset pitting and corrosion, in fact so that the boiler will last 20 years. Mr. Coryell seems to fear that these destructive forces will go on in spite of the several appliances used to prevent this evil. As many of the triple-expansion engines are now running without cylinder lubrication, and water losses made up by means of evaporators, the salt water and acids, alluded to as causing the pitting, will be conspicuous by their absence.

The statement that a Belleville boiler may be constructed of as light material and compare favorably in weight with other tubulous boilers, its durability being thereby lessened to their standard, would hardly seem correct.

Suppose we reduce the scantlings of the Belleville boiler *proper* to one-half their original thickness, which is certainly a large allowance, and see from Mr. Coryell's figures what result is obtained.

Reduction on elements, . . .	7,500 lbs.
" " separator, . .	550 "
" " sediment collector, . .	100 "
" " feed, . . .	100 "
" " water column, . .	350 "
" " piping, . .	200 "
Total reduction, . . .	8,800 "

Thus we have $\frac{35,430 - 8,800}{1,250} = 21.2$ as the weight per square

foot of heating surface. The corresponding weight for the Thornycroft boiler is 10.3 pounds, and for the Ward, 12.2 pounds. We find that even with this reduction the Belleville boiler is still approximately twice as heavy per square foot of heating surface as either the Thornycroft or Ward. This reduction has been made on a Belleville boiler with a ratio of heating surface to grate surface of 34 to 1, which has consequently made the reduction greater than it would have been had the ratio recommended of 26 to 1 been maintained. This supposed reduction would leave the Belleville boiler much weaker and less durable than either of the others.

I fail to understand Mr. Coryell when he states that it is impossible to increase the thickness of the tubes in these boilers *which have sacrificed safety and durability*. If we increase the thickness of these tubes to that of the Belleville tubes, we still have left an internal diameter of tube in the Thornycroft of $\frac{7}{8}$ inch and $1\frac{1}{8}$ inch, and in the Ward of $1\frac{1}{4}$ inch.

If this left too heavy a tube to expand, as in the Thornycroft, they could be readily fitted with threads in some such manner as in the Ward or other types of tubulous boiler. The circulation should cause no uneasiness, as tubulous boilers with smaller tubes than those given are in daily use. If Mr. Belleville is correct in his experiments, where he finds that up to 44 pounds of combustion per square foot of grate per hour the ratio of 26 to 1 may be relied upon, in his generator, to yield the same comparative vaporization per pound of coal, he has certainly arrived at results considerably at variance with practice here or in England.

The Belleville boiler of the yacht *Shearwater* gave on test an evaporation of 9.6 pounds of water per pound of coal, when burning 12.8 pounds per square foot of grate: had the combustion been forced to 44 pounds per square foot of grate, I seriously doubt if the same evaporation per pound of coal would have been maintained, basing my opinion on tests of other tubulous boilers under different rates of combustion. This is clearly shown in the table in the last number of the JOURNAL.

On so important a point as this, it is greatly to be regretted

(as well as somewhat singular) that no provision was made for forcing the fires in the *Shearwater* experiment, so as to prove this point.

For some reason Mr. Coryell has not maintained the ratio of 26 to 1 in his arrangement of Belleville boilers. The *Shearwater* had a grate area of 34.17 square feet, and a heating surface of 804 square feet, or a ratio of 23.5, while in the illustration the grate area is 37 square feet and heating surface 1,250, or a ratio of 34. Had the ratio recommended (26 to 1) been taken advantage of, we would have had 962 square feet of heating surface in each boiler, or 10,544 square feet in all.

I wonder if Mr. Coryell would be willing to guarantee with this heating surface equal results with 15,000 square feet in the Scotch boilers, at any rate of combustion under 44 pounds? If so, he could have reduced his head room sufficiently to have saved about 4,600 cubic feet in his arrangement.

[Owing to the lack of data in Mr. Coryell's remarks, the Secretary wrote to Messrs. Belleville & Co., requesting that they furnish particulars of some of their boilers, in order that they might have as thorough exposition in the consideration of Tubulous Boilers as the others. He is happy to state that Messrs. Belleville & Co. have forwarded a number of documents relative to tests, &c. Unfortunately these have come to hand too late to be translated and properly arranged for this number, but they will appear in the next.

It should also be stated that the Secretary repeatedly urged upon Mr. Coryell the importance of adding to his article data of trials and other actual performances of the Belleville boiler. He finally replied that he would endeavor to secure the data, but it is presumed that there was not sufficient time, as he has not sent them up to time of publication.—EDITOR.]

NOTES.

INSTRUCTIONS FOR DETERMINING THE COAL CONSUMPTION ON VESSELS OF THE FRENCH NAVY.

[Extracts taken from *Mittheilungen aus dem Gebiete des Seewesens*, 1890, No. 1.]

Since the method of determining the economic performance of the engine on the basis of miles made good is always accompanied with difficulties, it has been decided to use the number of revolutions of the engine as the basis.* In order to obtain data for the Regulation Table (*Tableau Type*), the Trial Board will make diagrams for each vessel, which will show the speed, the coal expended per hour, and the coal expended per nautical mile, as functions of the revolutions. Even the data of shortened or unfinished trial trips are to be used to make provisional tables.

Since the conditions under which trial trips take place are always very favorable, the Regulation Table contains, besides the columns for "coal expenditures on trial trips," the "permissible coal expenditure" per hour and per nautical mile on regular cruising, given as functions of the revolutions. The figures in the latter columns are 10 % greater than those in the former. This increase of 10 % was obtained from the data previously collected.

The actual coal consumption (h) for a given number of revolutions per minute, compared with the "permissible" consumption (h_1) shown by the table for the same revolutions, is intended to show, to a certain extent, the condition of the machinery, and

* NOTE BY TRANSLATOR.—Passed Assistant (now Chief) Engineer John Lowe, U. S. Navy, in a very good paper on the determination of the most economical speed for a steamship from two performances of that vessel (See *The Engineer*, January 2, 1880), said:

"Now the ratio between revolutions of the screw and the speed of the ship varies greatly with the weather; thus the wind may be so adverse that the revolutions may be great and the speed nothing. For this reason, the revolutions rather than the speed will be used as a basis for calculation."

the efficiency of the firemen and coal-heavers in managing the fires. For instance, if the machinery is in good condition, the co-efficient $\frac{h}{h_1}$ will give a fair measure of the efficiency with which the fires have been worked, and vice versa.

If m represents the actual number of miles made good with a given number of revolutions per minute, and m_1 the number of miles for the same revolutions according to the Regulation Table, the co-efficient $\frac{m}{m_1}$ will show the effect of outside conditions, (draught of vessel, cleanliness of hull, kind of weather, amount of sail carried, &c.,) on the speed of the vessel.

Speaking of the most economical speed, the article says that a practical method of determining this has not yet been found. The speed which is absolutely the most economical is, of course, the speed at which a mile can be made with the least expenditure of coal; but many engines are not fitted for this, so that their smallest performance would give only relatively the most economical speed. The instructions are, therefore, in general terms, to be as economical as possible with the fuel, considering the service on which the vessel is engaged. F. C. B.

SHIPS.

UNITED STATES.

Armored Cruiser No. 2.—Bids for the construction of this vessel (described on page 231 of the current volume) were opened June 10, with the following result:—

The Wm. Cramp and Sons' Ship and Engine Building Co., Department's designs throughout, $3,150,000; contractors' hull and Department's machinery, $2,985,000.

The Union Iron Works, of San Francisco, Department's design, $3,100,000; contractor's design, hull and machinery, $3,000,000.

The Risdon Iron Works, of San Francisco, Department's design, $3,450,000.

The contract was awarded to the firm of Cramp and Sons, on their second bid of $2,985,000 for their own hull and the Department's machinery.

Cruiser No. 6.—Bids for the construction of this vessel (described on page 230 of current volume) were also opened June 10. The only bidders were the Union Iron Works, of San Francisco, whose figures were $1,796,000 for the Department's design throughout, and $1,760,000 for their own design. The Department's design was considered enough better to warrant the award being made for it.

It may be interesting to note that in submitting the bids on their own designs for this vessel and *Armored Cruiser No. 2*, the Union Iron Works accompanied them in each case with complete detailed printed specifications, and blue-prints of hull and machinery in about the same detail as those furnished to bidders by the Department. It is said that the cost of these plans and specifications was nearly $10,000.

Gunboats 5 and 6.—(Described on page 351, Vol. I.) Bids were opened January 22d, with the following result:—

	For one vessel.	For both vessels.
Bath Iron Works, . . .	$327,000	$637,000
S. L. Moore & Sons, . . .	285,000	562,875
Atlantic Works, . . .	344,000	———

After some delay, Messrs. Moore & Sons withdrew their bid, and the contract was awarded to the Bath Iron Works, of Bath, Maine, for the Department's design throughout.

No work has as yet been done in the shops, but the detail drawings are being worked out and material ordered.

Naval Academy Practice Vessel.—(Described on page 352, Vol. I.) Bids were opened June 10, the bidders being F. W. Wheeler & Co., of Bay City, Michigan, $245,000, and Sam'l L. Moore & Sons, of Elizabethport, N. J., $250,000. Although Mr. Wheeler was the lowest bidder, it was found that the treaty of 1842 with England prevents the building of a war vessel on the

Lakes; consequently the contract has been awarded to the Messrs. Moore.

New Ships.—The recent Naval Appropriation Bill provides for three *Battle Ships* of about 9,000 tons displacement, with a speed of 15 knots; a *Protected Cruiser* of about 7,350 tons and 21 knots speed; a *Torpedo Cruiser* of about 750 tons and 23 knots; and a *Torpedo Boat.*

The *Battle Ships* will be the most powerful fighting ships in our Navy. The hull dimensions are 320 by 69 by 24 feet; displacement, 9,000 tons; immersed midship section, 1,530 square feet. There will be one military mast. The main battery will consist of four 13-inch, four 8-inch, and four 6-inch breech-loading rifles; and the secondary battery of twenty 6-pounder and four 1-pounder rapid-fire guns, besides two Gatling guns. There will be six torpedo tubes. The normal coal supply is 400 tons, and the bunker capacity 900 tons.

The machinery will consist of twin-screw, triple-expansion, vertical engines, of about 9,000 I.H.P. at about 128 revolutions per minute. The steam pressure is to be 160 pounds, and the air pressure at full power, 1 inch. The natural draft I.H.P. is to be 7,000. The cylinders are 34½, 48 and 75 inches diameter, by 42 inches stroke. Total condensing surface, 12,706 square feet.

There will be four double-end boilers, 18 feet long and 15 feet diameter, with 32 furnaces having a total grate surface of 552 square feet; the total heating surface is 17,463 square feet. The two auxiliary boilers are to be 8 feet 6 inches long, and 10 feet in diameter, with four furnaces containing 64 square feet of grate surface; heating surface, 1,937 square feet. For all boilers the grate surface is 616 square feet, and the heating surface 19,400 square feet.

The *Protected Cruiser* (*Cruiser No. 12*) will be, in some respects, the most interesting of our new ships. On account of the high speed and large I.H.P., it was decided, on the recommendation of Engineer-in-chief Melville, to fit triple screws, this being the first case of such an arrangement in our Navy, although the *Dupuy de Lôme* of the French Navy and the *Tripoli, Montebello* and *Monzambano* of the Italian Navy are so fitted.

The special advantages of triple screws are smaller size of separate parts of the machinery, enabling higher rotative speeds to be employed with safety, and greater economy in ordinary cruising, when one engine working up to about its natural draft full power will give a fair cruising speed with economy in fuel. It is far from economical to run large twin-screw engines designed for 19 knots and upwards at speeds of 9 and 10 knots. Engine friction alone forms such a large percentage of the power developed, to say nothing of the increased cylinder condensation due to slower rotative speeds, that the economic performance, as measured by coal per I.H.P., is very low. The slip of the single screw will, of course, be enormously greater, but it was shown years ago by Chief Engineer Isherwood's Mare Island screw experiments that a very large increase of slip may be attended with a very small reduction in total propelling efficiency.

The hull of this vessel is to have the following dimensions: 400 feet by 58 feet by 23 feet; displacement, 7,400 tons; immersed midship section, 1,169 square feet. Normal coal supply, 750 tons; bunker capacity, 2,000 tons. The main battery will be four 6-inch breech-loading rifles and eight 4-inch rapid-fire guns. The secondary battery will be twelve 6-pounder and six 1-pounder rapid-fire guns, besides two machine guns and one field gun. There will be six torpedo tubes.

The engines will be vertical, triple-expansion; cylinders, 42, 59 and 92 inches diameter by 42 inches stroke; revolutions at full power, 128; I.H.P. forced draft, 21,000; natural draft, 15,000. The boiler pressure will be 160 pounds, and the air pressure for full power, 1 inch. Total condensing surface, 28,371 square feet. Eight main boilers will be fitted, of which six are to be 21 feet 3 inches long and 15 feet 6 inches in diameter, and two 18 feet 8½ inches long and 11 feet 8 inches in diameter. Total number of furnaces, 56; grate surface, 1,221 square feet; heating surface, 41,559 square feet. The auxiliary boilers will be the same as for the *Battle Ships*, thus making the total grate surface 1,246 and the total heating surface 42,490 square feet. It is expected that the speed with full power on the trial will be about 22 knots.

The Torpedo Cruiser will have about the following dimensions: 260 feet by 26 feet 9 inches by 8 feet 4 inches; displacement, 750 tons. The I.H.P. will be about 6,000. The machinery has not yet been definitely settled, but will be either twin or triple screws. For the same reasons as in the new *Protected Cruiser*, triple screws would be preferable; but as weights must be cut down to the lowest point, it is possible that the slightly greater weight of the three engines, as well as the greater space necessary for them, will cause them to be given up. Coil boilers will, of course, be used. The success of the Ward boiler and its adoption for the *Monterey*, as well as its small weight for steam produced (as shown under *Monterey* in this number), would naturally lead to its adoption, and should the Government build the vessel this would doubtless be done. It is probable, however, that the contract will simply require coil boilers of a given weight and *approved* type, equal to the Ward, leaving the choice to the contractor, subject to the Department's approval.

The estimated weight of the hull alone is about 375 tons, and for the machinery about 250 tons.

The Torpedo Boat will probably be similar in most respects to the *Cushing*, which has proved a decided success. It is likely, however, that the machinery will be less complicated. It may be interesting to note that a famous English torpedo-boat builder, who inspected the *Cushing* very carefully, is said to have remarked that she has from two to three times as much machinery as he had ever found necessary to do the same work. It is certainly worth consideration whether the great regularity of crank effort due to the five cranks is not too dearly purchased by the increased number of parts to be adjusted and looked after.

Maine.—Work on the hull is so well advanced that it is intended to have the launch in November of this year. The outer plating is completed to the protective deck. There has been some delay in the receipt of material for this deck which has delayed other work.

The machinery is well advanced. All the cylinders have been cast, and several are finished and ready for erection. The

line and propeller shafting is finished, and the crank shafts are nearly done. The propellers have been cast and are nearly fitted. The boiler material is all on hand, and about one-third of the work on the boilers is done.

There has been a great deal of trouble with the steel castings for this vessel. Eighteen hollow cylindrical columns for the main engines have been rejected on account of shrinkage cracks, and the contractors, having requested the change, have been permitted to substitute hollow forged steel columns. Fifteen valve chest covers have also been rejected for failure under hydrostatic test. It is probable that cast-iron will be used instead of cast-steel for these covers.

Texas.—The machinery of this vessel is well along. The H.P. and I.P. cylinders are cast, and the machine work nearly done on them. The L.P. cylinder pattern is made and one cylinder nearly ready for casting. The condensers and boilers are about one-fourth completed. Some of the engine forgings have been received and are being machined. The auxiliaries and fittings are well in hand.

Work on the hull has been delayed a good deal in consequence of defective steel castings, which have had to be rejected. The framing is complete below the armor deck, and the outer plating is about half completed below that deck. The inner bottom is about nine-tenths completed. In general, about one-fourth of the work has been completed.

Philadelphia.—The contractor's forced-draught speed trial of this vessel took place June 25th, over a course of forty miles, previously laid out to the southward of Long Island and Block Island. The sea was smooth and the force of the wind from 2 to 3. The following are the mean results for the four hours :—

	Starboard.	Port.
Steam pressure in boilers, per gauge, .	159.8	
Steam pressure at engines, per gauge,	151.2	154
Steam pressure, first receiver (abs.), .	57.61	56.5
Steam pressure, second receiver (abs.),	24.52	24.3
Vacuum in condenser, . . .	24.3	24.67
Revolutions, . . .	119.58	119.55

		Starboard.	Port.
Cut-off in H.P. cylinders,		0.61	0.58
Mean pressures from indicator diagrams, but without corrections for errors of indicators,	H.P.	65.37	62.76
	I.P.	22.29	21.67
	L.P.	10.27	11.03
I.H.P. (uncorrected),	H.P.	1,765.3	1,704.9
I.H.P. (uncorrected),	I.P.	1,416.3	1,442.1
I.H.P. (uncorrected),	L.P.	1,440.9	1,549.4
I.H.P. Total,		4,622.5	4,696.4
I.H.P. Total, both engines,			9,318.90
I.H.P., air and circulating pumps,			115.06
I.H.P., auxiliary machinery,			178.69
I.H.P., total,			9,612.65

Average speed for 4 hours, 19.678 knots.

It was the intention to publish a fuller account of the trial in this issue, but, owing to a delay in correcting the indicators used, the final report on the performance of the machinery has just been received as we go to press. We add the corrected horse-power:

I.H.P. Total, both engines, about	8,533
I.H.P., air and circulating pumps, about	120
I.H.P., auxiliary machinery, about	162
I.H.P. total, about	8,815

A full account of the trial will appear in the November number of the JOURNAL.

San Francisco.—Work on this vessel is nearly completed, and preliminary trials have been made with very satisfactory results. On the 16th instant she ran for ten hours steadily, and for part of the time made 19½ knots, although in only seven fathoms of water. The contract trials will probably take place before the end of this month.

Newark.—The hull work is about nine-tenths completed, and will be finished in about five months. The machinery is practically complete, except fitting the propellers on the shafts.

Monterey.—The greatest interest attaching to this vessel is the use of coil boilers for about three-fourths of the power. It will be remembered that in August of 1888 an advertisement was issued by the Navy Department inviting manufacturers of coil boilers to submit plans and prices for boilers for 3,600 I.H.P. for the *Monterey*. This was subsequently changed to 4,200 I.H.P. After considerable delay, the competitive trials were ordered, only two makers having finally come forward to carry out the competition—Mr. Charles Ward and Mr. William Cowles.

Each boiler was given two trials, each lasting twelve hours under forced draft, with an air pressure of 2 inches of water. The Ward boiler used the closed fire room plan, and the Cowles the closed ash-pit. The complete report of the trials will be given in the annual report of the Engineer-in-Chief, but we are permitted to give the following abstract:

	Ward.	Cowles.
Date of tests,	Dec. 18 and 19, 1889.	Apr. 28 and May 8, 1890.
Boilers tested:		
Grate surface, square feet,	53	47
Heating surface, square feet,	2,473.5	2,026.75
Weight of empty boiler without smoke-pipe, tons,	11.84	9.75
Weight of same with water, tons,	13.85	11.55
Ratio H.S. to G.S.,	46.67	43.12
Proposed Boilers:		
Number for the *Monterey*,	4	6
G.S. of each,	75	47
H.S. of each,	2,938	1,998.5
Weight empty, tons,	13.58	9.6
Weight with water, tons,	15.86	12.65
Duration of test, hours,	24	24.15
Fuel consumed, total, pounds,	70,022	45,620
Refuse from fuel, total, pounds,	3,389	6,327
Combustible consumed, pounds,	66,633	39,293
Per cent. of refuse,	4.84	13.87

	Ward.	Cowles.
Total feed water used, pounds,	461,885	280,822
Temperature of feed water,	50.4	58
Steam pressure, pounds,	160	160
Air pressure in inches of water,	2	2
Coal per hour per sq. ft. of G.S.,	55.05	40.19
Combustible per hour per sq. ft. of G.S.,	52.4	34.62
Coal per hour per sq. ft. of H.S.,	1.18	0.932
Per cent. of moisture in steam,	11.62 13.35	
Per cent. of superheating,		7.5 3.8
Apparent evap'n from temp. of feed at temp. of steam, per lb. of coal,	6.60	6.16
Same per pound of combustible,	6.93	7.15
Potential evap'n (allowance being made for moisture or superheating), under above conditions per lb. of coal	6.00	6.50
Same per pound of combustible,	6.30	7.54

These tests were made in the usual way for all evaporative tests, starting with steam raised and bare grates, and ending with no fire and steam not forming. This, of course, is the accurate way of getting the evaporation, but it does not give a fair idea of what the boiler will do for a number of hours at full power. To show this feature, ten consecutive hours have been taken for each boiler, and, in addition, a correction has been made for the feed water at 120° instead of the temperatures during the trials.

MAXIMUM PERFORMANCE FOR TEN HOURS.

	Ward.	Cowles.
Apparent evaporation per hour from feed temperature of 120° at 160 pounds pressure,	22,050	13,200
Actual evaporation under above conditions (correction made for moisture and superheating),	19,105	14,192

	Ward.	Cowles.
Actual evaporation per hour per square foot of heating surface, . . .	7.724	7.002
Horse-power for which one of the boilers proposed for the *Monterey* will furnish steam (based on ratio of heating surfaces) at 20 pounds of steam per I.H.P. per hour, ...	1,139.5	709.6
Horse-power from the whole number of proposed boilers, ...	4,558	4,257.6

In the decision as to which boiler should be used in the *Monterey*, account was taken not only of the relative evaporation per square foot of heating surface, but of the relative weight for power developed, space occupied, facility for repairs, convenience of manipulation, &c., &c. Without going into detail, it may be said, as an inspection of the tables will show, that in most of these items the Ward boiler came out ahead. It was accordingly recommended that this boiler be used in the *Monterey*, and a contract has recently been made with Mr. Ward for four of his boilers of the size proposed by him.

The contract price, which includes transportation to San Francisco, erection on the vessel, insurance, spare parts, &c., is about $45,000.

The hull is about one-fourth completed, and it is expected that so far as it is concerned the vessel can be ready for trial in October, 1891.

Cruisers 7 and 8.—The machinery for these vessels is building at the New York Navy Yard, and work on it is progressing rapidly. Of the sixteen main cylinders, thirteen have thus far been cast without a single failure, and the other three are now in the foundry. A great many of the small castings have been made and are being machined. The connecting rods are on hand and being machined. Work on the boilers is in a backward condition on account of having to wait for the erection of hydraulic flanging plant, but will soon be pushed. Considerable trouble has developed in getting the engine columns. These are

of cast steel, and, after numerous attempts, the contractors report that they cannot make them. The trouble appears to be due to the cores, which cannot be removed in time to prevent shrinkage cracks. Other makers of steel castings have succeeded with similar forms, however, and it is likely that the columns can be procured from them if the present contractors are finally unsuccessful.

The hull work on these vessels is about 15 per cent. completed. Some delay has been caused by the rejection of defective steel castings. *Cruiser No. 7* is building at the New York Yard and *Cruiser No. 8* at the Norfolk Yard.

Cruisers Nos. 9 and 10.—The keels and stems are in place, also much of the framing and some outside plating. Most of the hull material is on hand.

Work on the machinery is progressing favorably. Most of the main cylinders, pistons and other heavy castings have been made and partly machined. Some of the engine forgings have been received and are being finished. Work on the boilers is well in hand.

The contractors have received five payments (one-twentieth of contract price each) thus indicating that, including labor and material, they are about one-fourth completed.

Cruiser No. 11.—Very little work has been done thus far on the hull, except procuring material. The yard of Mr. H. Loring has been fitted out for good work, and future progress will be rapid. The machinery is building at the Quintard Iron Works, and is much further along. The main cylinders are cast, a good deal of work done in the pattern and blacksmith shops, and most of the boiler plates have been received and pickled.

Concord.—The contract work on this vessel is almost completed, and the preliminary trials will take place this month.

Bennington.—Work on the machinery is completed, except some of the lagging and a few fittings.

Vesuvius.—Since the notice of the horse-power trial of this vessel in the last number of the JOURNAL (pp. 231–2), the horse-power has been computed, with the following result:—

Starboard engine: steam pressure, 154 pounds; first receiver, 72.5 pounds abs.; second receiver, 28.2 pounds abs.; vacuum, 24.6 inches; revolutions per minute, 267.78.

	Mean pressure.	I.H.P.
H.P. cylinder,	51.22	496.20
I.P. cylinder,	30.61	620.82
Forward L.P. cylinder,	15.46	374.01
After L.P. cylinder,	16.70	403.91

Aggregate mean pressure reduced to L.P. cylinders, 39.04; aggregate I.H.P., 1,894.94.

Port engine: steam pressure, 160 pounds; first receiver, 77.7 pounds abs.; second receiver, 29.1 pounds abs.; vacuum, 24.5 inches; revolutions per minute, 270.

	Mean pressure.	I.H.P.
H.P. cylinder,	56.03	547.37
I.P. cylinder,	26.20	535.83
Forward L.P. cylinder,	14.15	345.03
After L.P. cylinder,	15.94	388.69

Aggregate mean pressure reduced to L.P. cylinders, 37.14; aggregate I.H.P., 1,816.92.

The I.H.P. of the auxiliaries was estimated as follows:—

Blowers, 50.00; circulating pump, 20.00; feed pump, 12.00; bilge pump, 1.00—making the aggregate for the auxiliaries 83.00 I.H.P.

The total I.H.P. of all machinery was thus 3,794.86.

It is greatly to be regretted that no effort was made, even with a patent log, to get some idea of the speed.

As the trial was of very short duration, only eighteen minutes, it was important to get as many indicator cards as possible from each cylinder to ascertain whether the power was uniformly maintained, and to permit of rejecting defective cards without losing the record of any cylinder. There was but one indicator to each cylinder, with a connection to each end. The two sets of engines are side by side, and the observers were on a narrow platform between them, each observer taking cards from the corresponding cylinder of each engine. As showing how quickly such work can be done, Assistant Engineer Albert Moritz se-

cured nine perfect cards from each after low-pressure cylinder, besides several that were slightly defective. This included removing and fitting the cards, but not the recording of any data except the number of the card and the time.

For the benefit of those readers of the JOURNAL who have not the number, (No. 1, Vol. 1,) which contained the description of this vessel and its machinery, the following dimensions are given:—

Hull: length, 246 feet 3 inches, beam, 26 feet 6 inches; mean draught, 9 feet 3 inches; displacement at above draught, 805 tons. Machinery: twin-screw, vertical, triple-expansion engines, each having one H.P. cylinder 21½ inches, one I.P. cylinder 31 inches, and two L.P. cylinders each 34 inches in diameter by 20 inches stroke, both engines being in the same compartment. The air pumps are worked off beams from the L.P. cylinders; the circulating pump is independent. There is one condenser for both engines. The boilers are four in number, of the low cylindrical (sometimes called marine-locomotive) type, 9 feet in diameter and 19 feet 8 inches long. (For the other data of machinery, see p. 232 of current volume.)

On her successful speed trial, consisting of two runs in opposite directions over a course 2.5 knots long, the mean speed was 21.65 knots.

It is stated that the average revolutions were 271.85, the boiler pressure 160 pounds, and the aggregate I.H.P. 4,295 for the main engines only. As the engine data were not taken under government supervision, and the indicators were not standardized, the actual I.H.P. developed is not known. Judging from the results of the horse-power trial, however, it is probable that the I.H.P. above is not more than five per cent. in excess of the true I.H.P.

ENGLAND.

Blenheim.—This splendid vessel, the largest protected cruiser afloat, was launched July 5th, from the yard of the Thames Iron Works and Shipbuilding Co. Her dimensions are 375 feet by 65 feet by 25 feet 9 inches, with a displacement of 9,000 tons.

"This is a greater displacement than has ever been previously reached by a cruiser, but it is justified by the fact that the *Blenheim* will have engines of 20,000 I.H.P. under forced draft, giving a speed of 22 knots. The engines will be supplied by Humphreys, Tennant & Co. Four thousand tons have been assigned to the weights for protection, armament, equipment and coal. The *Dupuy de Lôme*, the crack French cruiser, has vertical armor 4 inches thick running from stem to stern, but in the *Blenheim* protection is afforded by a steel deck from 3 to 6 inches thick, covering the whole vessel and efficiently protecting the machinery and magazines. The armament will consist of two 22-ton and ten 6-ton breech-loading rifles; sixteen 3-pounder rapid-firing guns; one 1-inch and seven 0.45-inch Nordenfeldt guns, and four 14-inch Whitehead torpedo tubes."

Medusa Class.—These vessels have already been described in Volume I. In his article, "Notes on Recent Naval Manœuvres," Mr. W. H. White called attention to the marked influence of even a slightly foul bottom on the speed. The *Medea*, owing to delay, was tried with a foul bottom and required 5,800 I.H.P. for 16.9 knots, while the sister vessel, *Medusa*, made 17.6 knots for this I.H.P., and 17 knots on 5,000 I.H.P. For 13 knots the *Medea* required nearly 2,000 I.H.P. as against 1,600 for the *Medusa*. The cost of the hulls of these vessels was £88,400, and of the machinery £53,300.

Latona.—The first of twenty-nine similar vessels, was launched at the works of the Naval Construction and Armaments Co., Barrow-in-Furness, May 22. Twelve of these vessels are to be built in the government dockyards, and the rest by private firms, the *Latona* being the first of three building at Barrow. Length, 300 feet; beam, 43 feet; mean draught, 16 feet 6 inches; displacement, 3,400 tons. The engines are of the usual twin-screw, triple-expansion type, and vertical, being very light, as in all war vessels. The cylinders are 32½, 49 and 74 inches diameter by 36 inches stroke, designed for 9,000 I.H.P. with moderate forced draft, and 7,000 I.H.P. with natural draft. The boilers have an aggregate heating surface of 16,000 square feet. A protective deck is fitted, and in addition the engines above this deck are

enclosed in a belt of 5-inch armor with 7-inch teak backing around the engine hatch, extending from the protective to the upper deck. The engines are in separate water-tight compartments, and the boilers are grouped in two water-tight compartments, each group having a separate smoke-pipe.

The battery consists of two 6-inch breech-loading rifles, with central-pivot carriages, one on poop and one on forecastle; six rapid-firing 4.7-inch guns, three on each broadside; eight rapid-firing 6-pounder guns, four on each broadside, besides one 3-pounder Hotchkiss and four 5-barreled Nordenfeldt guns. Four torpedo tubes are fitted, one forward, one aft, and one on each broadside forward of the poop.

There are two pole masts, with light fore and aft rig. The crew will be about 250, all told.

The sister ships to the *Latona*, building at Barrow, are the *Naiad* and *Melampus*.

As will be seen at a glance, these vessels are an enlarged edition of the *M* class, with less powerful machinery. They ought to be more satisfactory vessels in every way, and will probably not disappoint the expectation of 19¾ knots under forced draft.

Aeolus, *Brilliant*, *Sybille*.—These vessels are similar to the *Latona*, having the same length and beam, but a foot more draught (17 feet 6 inches) and 200 tons greater displacement (3,600). The hulls, designed by Mr. W. H. White, are building in the government yards. The machinery, by Hawthorn, Leslie & Co., consists of twin-screw, triple-expansion, vertical engines, with cylinders 32½, 49 and 74 inches diameter by 39 inches stroke. There will be three double-end and two single-end boilers, working at 160 pounds pressure. The I.H.P. expected is 9,000 with forced draft, giving 19¾ knots, and 7,000 with assisted natural draft, giving 18¼ knots.

Plassy.—A swift torpedo gunboat for the Indian government was launched at Elswick, July 5. She is 230 feet long, with a displacement of 735 tons. Machinery of 4,500 I.H.P. will be fitted, which is expected to give a speed of 21 knots.

Barham.—A second attempt was made July 9 to complete her

trials. The defects developed on the first trial had been corrected, and the changes answered their purpose admirably, but otherwise the trial was a failure. "During the third hour, when a speed of 17 knots had been reached with an air pressure of one inch, and when the engines were exerting more than the contract power, the tubes of several of the boilers began to leak and the ship returned to port. The *Barham* will not be ready for another trial for some time." One of the changes made was lengthening the smoke-pipe several feet.

Ringdove.—An eight-hour natural-draft trial took place July 17. On a previous trial, some months before, the boiler tubes leaked badly, so that new ones were fitted. The last trial was successful. Steam pressure, 135 pounds; revolutions, 167; vacuum, 26½ inches; I.H.P., 822; speed, 12½ to 13 knots. The contract I.H.P. with natural draft was 720. There were no leaky tubes this trial.

FRANCE.

Charner, *Bruix*, *Chanzy* and *Latouche-Tréville.*—The following additional information is given for these vessels (already described on pages 235–6 of current volume): Weight of hull 2,813.6 tons; weight of machinery and contained water, 702 tons. They are expected to make 17 knots with about 6,600 I.H.P. under natural draft, and 19 knots with about 8,200 I.H.P. under forced draft. The coal supply at normal displacement is 394 tons, giving a radius of action at 10 knots of 4,000 knots. The propellers are 14 feet 3 inches in diameter and have a pitch of 16 feet 6¾ inches. The estimated cost of the *Charner* and *Bruix*, which are to be built at Rochefort, is $1,798,000. The contract price for the other two, building by private firms, is $1,236,000 for the hull and $400,000 for the machinery. The engines have Marshall (Bremme) valve gear.

Tréhouart.—A sister ship to the *Bouvines*, *Valmy* and *Jemmapes* (described on page 235), is to be built by the Government, the hull at Lorient and the machinery at Indret. Belleville boilers will be used in this ship instead of the d'Allest boilers which are to be fitted in the others. The estimated cost of the *Tréhouart*

is $2,780,000. The following information in regard to these vessels is additional to that previously given: Grate surface, 645.8 square feet; heating surface, 21,095 square feet; ratio, 32.7 to 1. The coal consumption under natural draft (based on an eight-hour trial) must not exceed 2.25 pounds. Under these conditions, the expected speed is 16 knots, with about 100 revolutions per minute. Under forced draft, the combustion must be at least 30.75 pounds per square foot of G.S. (150 Kg. per square meter). The stroke of the engines is to be one meter (39.37 inches). The diameters of cylinders may differ somewhat for the different ships, as the sizes have been left to the contractors. The propellers will be three-bladed and of manganese bronze.

ITALY.

Andrea Doria.—This battle ship, of the same class as the *Ruggiero di Lauria* and *Francesco Morosini*, had her full-power steam trials July 15 off Spezzia. The hull dimensions are 328 feet by 65 feet 4 inches by 27 feet 1 inch; displacement, 11,000 tons. The machinery, by Maudslay, Sons & Field, was designed for 10,000 I.H.P. The engines are vertical, twin-screw and triple-expansion, fitted with Joy's valve gear. The mean results for the trial were 10,500 I.H.P. and 16.1 knots. The air pressure was only ¾ of an inch.

Partenope.—One of a class of torpedo-catchers designed by the Italian Navy Department. Length, 246 feet 2 inches; beam, 24 feet 7 inches; displacement, fully equipped, 826.7 tons. The hull of this boat was built at Castellamare. The machinery for the whole class is to be built by Schichau. There are two sets of triple-expansion engines, supplied with steam by four locomotive boilers, working at 180 pounds pressure. The forced draft is by Schichau's system of closed ash-pits.

The contract conditions were that the engines should indicate 4,000 I.H.P. (metric) under forced draft during a continuous run of three hours with a maximum coal consumption of about 2.5 pounds per I.H.P. per hour. Under natural draft the I.H.P.

was to be 2,000 (metric) for twenty-four hours continuously, with a maximum coal consumption of 2 pounds per I.H.P. per hour. Premiums and penalties were provided in connection with these requirements.

The trials took place in March, with excellent results. The I.H.P. was 4,150 to 4,200 (metric), say 4,125 in English units, with an air pressure from 1½ to 2 inches of water, and the coal consumption is said to have been 2.1 pounds per I.H.P. per hour. The natural draft trial lasted for twenty-four hours, giving about 2,175 I.H.P., with an alleged coal consumption of about 1.5 pounds per I.H.P.

RUSSIA.

Emperor Nicholas I.—This armored vessel was commenced in March, 1886, and launched in May of last year. Length over all, 346 feet; length between perpendiculars, 331 feet 6 inches; beam, 67 feet 1 inch; mean draught, 23 feet; displacement, 8,310 tons. The engines are twin-screw, vertical, triple expansion, and arranged like those of the *Maine*, with the L.P. cylinders forward, provision being made for disconnecting them and using the other two cylinders as a compound engine at ordinary cruising speeds. The cylinders are 39, 57 and 83.14 inches diameter by 38⅝ inches stroke. The cooling surface in both condensers is 13,500 square feet. The propellers are four-bladed, of phosphor bronze. The contract I.H.P. is about 7,900. There are twelve cylindrical boilers, 13 feet 2 inches diameter and 9 feet 10 inches long, each having three furnaces. The total grate surface is 816 square feet, and the total heating surface 19,393 square feet. Working pressure, 125 pounds. The weight of machinery, including water, is about 1,080 tons. The cost of the machinery is to be $1,030,000, and of the hull, $2,197,792.

Sinope.—An armored battle ship, had her steam trials on May 7. She is similar to the *Tchesme* and *Catherine II*, except that she has triple-expansion engines. The hull is 339 feet 6 inches by 69 feet by 26 feet 6 inches, with a displacement of 10,150 tons, and was built at Sevastopol. The machinery was built by

Napier & Sons, of Glasgow. The engines are vertical, twin-screw in separate compartments, with a longitudinal water-tight bulkhead between them, which also extends throughout the boiler compartments. The cylinders are 44, 63 and 95 inches diameter by 45 inches stroke, and fitted with Kirk's valve gear. The working pressure is 125 pounds. There are 14 single-end boilers, ranged with their backs against the central longitudinal bulkhead, the fire room being outboard. Transverse bulkheads divide the boilers into four separate groups. The boilers are 15 feet 1 inch diameter and 10 feet 6 inches long, each having three furnaces. Total grate surface, 1,064 square feet; total heating surface, 26,222 square feet. Steam jets are fitted for aiding the draft. The shafting is of steel, the crank shafts being 18 inches diameter. There is one piston valve to each cylinder, with arrangements for supporting its own weight, the lower piston being larger than the upper. There are two air pumps to each engine, worked from the H.P. and I.P. cross-heads. The centrifugal circulating pumps are independent.

The doors in water-tight bulkheads are fitted in a somewhat novel way, being above the floors and reached by short ladders. If a compartment is flooded, this gives time to close the door before the water can get into the adjoining compartment.

The mean I.H.P. and speed on the trial were 12,803 and 17 knots.

A subsequent trial for economy is said to have shown the development of a horse-power for 1.35 pounds of coal, but it is needless to say that no reliance can be placed on the accuracy of such figures.

Ruric.—A battle ship of about 11,000 tons displacement, had the keel laid recently. The dimensions of the hull are 426 by 67 by 27 feet. The expected speed is 18 knots. The side armor is 10 inches thick.

Navarino.—An armored turret ship, whose keel was laid at the same time as that of the *Ruric*. Hull dimensions, 360 by 68 by 25 feet; displacement, about 10,000 tons. The I.H.P. is expected to be 9,000 from twin-screw engines and twelve boilers, giving a speed of 17 knots.

Torpedo Boats.—Three new torpedo boats, built by Schichau for the Russian government, were tried in May. The torpedo cruiser *Lieutenant Kasarsky* is 190 feet 3 inches long, 24 feet 7 inches beam, and has a displacement of about 400 tons. The engines are twin-screw, triple-expansion of about 3,500 I.H.P. There are two locomotive boilers, with Schichau's forced draft arrangement, working at 180 pounds. The trial lasted three hours, all weights being on board, and in addition the bunkers were full (95 tons). A mean speed of 21 knots was easily maintained without unduly forcing the engines.

The torpedo boat *Anakreon* is 128 feet long, 19 feet 8 inches beam, and about 85 tons displacement. The contract speed was 21 knots, and this was easily maintained for two hours.

The torpedo-despatch boat *Adler* is of a new type introduced by Schichau, and is similar in principle to the Italian boats *Aquila*, *Nibbio* and *Falko*. The hull is 152 feet 7 inches long, 17 feet beam, and about 150 tons displacement. The engines are twin-screw, triple-expansion of about 2,300 I.H.P. There are two locomotive boilers. The mean speed for a two-hours' run was 26.55 knots, and the mean of six runs over the measured mile was 27.4 knots, making this little boat the fastest in the world.

HOLLAND.

Prinses Wilhelmina der Nederlanden.—A protected cruiser, with a steel hull, sheathed with pitch-pine. The armor deck extends throughout the whole length, and there is an armored turret forward. The stem, stern-post and rudder-post are of manganese bronze, which is to have a tensile strength of about 56,600 pounds per square inch, with an elongation of 6 to 7 per cent. A belt of cellulose fibre is fitted at the sides from the armor deck to some distance above the water-line. There are ten water-tight compartments below the armor deck and seven above it, with a longitudinal bulkhead between the engines. The hull dimensions are 328 feet by 48 feet 10½ inches by 19 feet 8 inches; displacement, 4,530 tons. This is fully equipped, including 440 tons of coal.

The engines are vertical, triple-expansion, driving twin-screws. There are four sets, two to each shaft. The I.H.P. with forced draft (air pressure about 2 inches) is expected to be about 5,825.

The cylinders are 19, 31, and 51 inches diameter by 27 inches stroke. Condensing surface, 8,000 square feet in four copper condensers. The shafting is of hollow steel. The propellers are of Delta metal, 11 feet diameter, with a pitch variable from 11 to 14 feet. The four circulating pumps have a combined capacity of about 440 tons per hour. The expected revolutions of main engines are 170 per minute, which is intended to give the ship, with a displacement of about 4,330 tons, a speed of 16½ knots per hour.

There are four cylindrical boilers for a working pressure of 170 pounds. Grate surface, 280 square feet; heating surface, 10,000 square feet; ratio, 35.7 to 1. Steam space in boilers, 1,400 cubic feet; diameter of smoke-pipe, 8 feet.

The forced draft will be by closed fire rooms, the fans drawing air from the adjoining compartments as well as the outside air. An evaporator is provided for "make-up" in the feed-water.

The battery consists of one 30-caliber 28-centimeter (11.03 inches) breech-loading rifle in the turret, with an angle of train of 150° on each side; one 35-caliber 21-centimeter (8.27 inches) breech-loading rifle on after deck, with an armored shield and about the same train as the forward gun; two 35-caliber 17-centimeter (6.69 inches) breech-loading rifles, one on each side amidships in towers, so as to fire directly ahead or astern. The secondary battery consists of six machine and eight rapid-firing guns on the superstructure and in the tops. There are also two torpedo guns on each side and one torpedo tube at the bow.

A complete electric installation is fitted with search lights and incandescent lamps. There are 220 of the latter fed from two dynamos, each furnishing 150 amperes at 70 volts, in connection with storage batteries.

BRAZIL.

Almirante Tamandaré.—A protected cruiser, rigged as a frigate, with 17,350 square feet of sail area. The hull is 294 feet 2 inches

by 47 feet 4 inches by 19 feet 9 inches; displacement, 4,662 tons. The engines are horizontal, twin-screw, triple-expansion, in separate compartments. The cylinders are 36¼, 55 and 84 inches diameter by 39 inches stroke. There are seven boilers with 28 furnaces; working pressure, 130 pounds. I.H.P., 6,400 with natural draft and 7,400 with forced draft. Coal capacity, 714 tons.

The hull was built in the dockyard at Rio de Janeiro, and launched March 20, 1890, having been over six years in building. It is of steel and sheathed. The machinery is by Maudslay, Sons & Field. The total cost of the vessel is expected to be about $2,017,000.

ARGENTINE REPUBLIC.

Necochea.—A fast protected cruiser, somewhat like the *Piemonte*, was launched at Elswick, May 5. She is 300 feet long and 43 feet beam, with a displacement of 3,200 tons. The I.H.P. is to be 13,500, which is designed to give 22 knots, but it is thought this speed may be exceeded on trial.

JAPAN.

Chiyoda.—A belted cruiser, built for the Japanese government by Messrs. J. and G. Thomson, and launched June 3d. She is somewhat similar to the Colonial cruisers, *Tanranga* and *Ringarooma*, building by Messrs. Thomson, but is about 45 feet longer, giving an advantage for high speed. The hull is 310 by 42 by 14 feet, with a displacement of 2,450 tons. The expected speed at full power under forced draft is 19 knots.

The machinery consists of twin-screw, vertical, triple-expansion engines, with cylinders 26, 39 and 57 inches diameter by 27 inches stroke. The propellers are of manganese bronze, with adjustable blades. There will be six locomotive boilers, in two separate compartments.

The water-line, for about two-thirds of its length amidships, will be protected by a belt of rolled steel plates secured to the outside of the shell plating, its thickness being 4⅜ inches. There will also be a 1-inch steel deck over the machinery and maga-

zines, and extending throughout the whole length. This deck is in two thicknesses, the lower being the usual mild steel, and the upper chrome steel, on account of its greater resistance to penetration.

The main armament will consist of ten 4.7-inch Armstrong quick-firing guns. The secondary battery will be fourteen Hotchkiss 47 mm. rapid-fire guns and three Gatlings mounted in the tops. There will be three torpedo tubes for Schwartzkopf torpedoes, one at the bow and one on each broadside.

Matsusima and *Itsukusima.*—These protected cruisers are building for the Japanese government in France. The former was launched at La Seyne, January 22, 1890, the keel having been laid in November, 1887. Length, 324 feet 9 inches; beam, 51 feet; draught aft, 21 feet 2 inches; displacement, 4,210 tons. The engines are twin-screw, with a combined I.H.P. of about 5,330. The expected speed is 16 knots.

MERCHANT STEAMERS.

Seguranca.—This vessel, the latest addition to the United States and Brazil Steamship Line, was launched from Roach's shipyard at Chester, in May. Length over all, 336 feet; beam, 45 feet 6 inches; depth under spar deck, 27 feet 9 inches; dead weight capacity, 3,800 tons; displacement, 4,890 tons. The engines are 30, 46 and 74 inches diameter by 48 inches stroke. There are six boilers with corrugated furnaces working at 160 pounds pressure. The expected speed is from 16 to 17 knots.

City of Paris.—All the circumstances connected with the accident to this vessel (pp. 238–244 current volume) have been investigated under the direction of the Board of Trade. During the investigation, some of the most eminent marine engineers and naval architects of England were called as experts.

After an exhaustive discussion, the court delivered an opinion, from which we extract the following in regard to the cause of the accident:

"We are of opinion that the primary cause of the casualty was the extraordinary wearing down of the bearing of the bracket which supported the extreme end of the propeller shaft, by which the end had dropped down from its original and proper position about seven inches; that this bearing down of the after end, its distance (about 50 feet) from its nearest support in the stern tube, coupled with the weight of the propeller and of the shaft itself, produced a bending effect on the shaft and its forward support, co-existent with each revolution of the engine; that this had probably produced rupture of the external surface, which gradually extended inwardly, and finally to total fracture of the shaft; that the engine, being relieved from all resistance beyond the friction of its own moving parts, would almost instantaneously have acquired a greatly accelerated velocity, the result of which was probably sufficient, by the additional strains on the various connections, to diminish the clearance allowed between the piston and cylinder cover of the low-pressure engine at the top end of its stroke; that the cover was struck by the piston and moved off, the piston probably broken up in the act, the cylinder torn asunder, the condenser seriously broken, and the destruction of the various other parts rapidly accomplished.

"As to the cause of the wearing down of the after bearing of the propeller shaft, considerable difference of opinion appeared to exist. This wearing was of an abnormal character and of unfrequent occurrence. Generally, it may be said to have been attributed to two distinct causes, one being that the gun-metal casing or lining of the shaft had burst through, being in a state of excessive tension; the other cause suggested was that ashes, which had been more frequently thrown overboard on the starboard side than the port, had got into the bearing and set up the action that led to this wear. But there is scarcely a steamship running whose outer bearings are not subject to the conditions that may be supposed to arise through ashes or other

debris from the ship. What seems to have occurred is that, from some cause or other, the waterways in the lignum vitae strips became choked, or were insufficient for the passage of water; that the friction which was generated soon destroyed the lignum vitae; that the metal casing of the shaft and bush being brought in contact, heat was further developed sufficient to break up the metal casing of the shaft, the fragments of which were found in the outer protecting case of the shaft, presenting the appearance of metal that had been broken while hot; and that the high temperature of the rubbing surfaces of the shaft and its bearing caused the extraordinary wear that followed."

It is to be noted that this explanation of the wrecking of the engine is by no means generally accepted, and that several letters to the technical papers have pointed out that the supposition of the piston striking the cylinder cover is inconsistent with the position and condition of the parts of the wrecked engine after the accident and of the strength and properties of the materials, as well as the testimony of the people in the engine room when the accident occurred.

It is pointed out that to suppose the striking of the cylinder head involves a stretching of the connecting and piston rods, and the cap bolts of main journals, that is incredible. On the whole, it appears to be one of those cases where even the best informed may differ in the absence of conclusive data.

The opinion of the Court is very interesting in another particular: "We may observe that during the course of this inquiry we had evidence that, even if some of the forward compartments of this ship had been filled with water in addition to those already filled aft, she would still have had a fair amount of freeboard, and have been able to float under the circumstances, her mean draught of water on arrival at Queenstown being only 8 or 9 inches more than when she left New York."

Teutonic.—This vessel now holds the record for speed in the transatlantic race, having completed, August 13th, the fastest westward voyage ever made, in 5 days 19 hours and 5 minutes. Prior to this the *City of Paris* had made the best time, 5 days 19

hours and 18 minutes, which was done last year. The *City of New York* left Queenstown about an hour after the *Teutonic*, and the two raced all the way to New York, but the former was beaten by about three hours. The daily runs of the *Teutonic* were 473, 493, 512, 500, 485 and 340 nautical miles; and those of the *City of New York* 464, 486, 485, 494, 474 and 388 nautical miles.

La Touraine.—This fine vessel, the latest and largest of La Compagnie Générale Transatlantique (briefly described on p. 362, vol. 1), has recently been launched. The following additional details are from *le Génie Civil:* Length, 516 feet 6 inches; beam, 55 feet 11¼ inches; mean draught, 23 feet 7½ inches; displacement, 11,490 tons. The engines are twin-screw, vertical and triple-expansion. The cylinders are 40.94, 60.63 and 100 inches diameter by 65.35 inches stroke. The piston speed is about 790 feet per minute, or 72 revolutions. The I.H.P. of each engine is about 6,250 (metric). Cast steel is used for the condensers and engine framing. The moving parts are all of forged steel, and only the cylinders are of cast iron. All the shafting is of steel and hollow. The crank-shaft journals have external and internal diameters of 20.3 and 6.3 inches. The propellers are three-bladed, with the blades of bronze and the hub of cast steel. They are 19 feet 8 inches in diameter. The engines are separated by a longitudinal water-tight bulkhead.

There are six double-ended and three single-ended boilers 14 feet 9¼ inches in diameter. The double-ended boilers are 20 feet 4⅛ inches, and the single-ended 10 feet 2 inches long. There are in all 45 corrugated furnaces. Moderate forced draft by closed ash-pits (Audenet's system) is used. The working pressure is 150 pounds.

The expected speed on trial is 19.5 knots, and at sea in ordinary weather 18.5 knots.